Using Sensor-based Demand Controlled Ventilation to Realize Energy Savings in Laboratories

THESIS

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Abstract

The building sector in the United States accounted for 41% of domestic and 7% of global energy consumption in 2010, with heating, ventilating, and air-conditioning (HVAC) activities consuming approximately 41.4% of the total facility energy consumption. Within the HVAC system, the parasitic energy accounts for one-third of the total energy consumed while heating and cooling accounts for the balance. The fan energy is approximately 85% of the total parasitic energy in the HVAC system. In a laboratory, energy related to ventilation can account for nearly half of the electrical energy demand. A carbon dioxide (CO_2) – based demand controlled ventilation (DCV) strategy can reduce the ventilation requirement by monitoring the indoor air quality (IAQ) of a space and modulating the ventilation based on the real-time occupancy.

This research presents a tool for laboratory managers to quickly determine if employing a DCV system is potentially life-cycle cost effective. The tool presented is not to be used as sole justification for implementing a DCV system; instead, laboratory managers using this tool will be able to quickly determine if further investigation into DCV installation is warranted. The results show that a DCV system is life-cycle cost effective for many different HVAC system total pressure and square footage combinations.

To my Wife and Daughter, I love you.
Thank you for all of your love and support.

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I. Introduction

Two factors affecting energy demand are population growth and energy use per capita (Reddy, 2000). "The world population has increased explosively over the past 100 years" (Reddy, 2000, p. 50) and is expected to continue increasing. This growth is going to place increased stress on all aspects of the global energy system. "In fact, 49 percent of the growth in world energy demand from 1890–1990 was due to population growth, with the remaining 51 percent due to increasing energy use per capita" (Reddy, 2000, p. 51). Neither factor is expected to decrease, resulting in an ever-increasing energy demand. Rising energy demand throughout the world has significant negative impacts.

The negative impacts of this increasing energy requirement are greater demands on the energy system, specifically on fossil fuels, and climate change. Analysis of the world energy supply at the current rate of consumption indicates that the current system is unsustainable and will have lasting impacts into the future (UNDP, 2000). The energy system is also partially responsible for global climate change due to the release of greenhouse gases and chlorofluorocarbons (Holdren & Smith, 2000).

Two ways to address rising energy consumption and the resulting negative consequences are to improve energy efficiency and reduce energy demand. Significant achievements can be made to improve energy efficiency because approximately two-thirds of energy is lost in the conversion from primary to useful energy (UNDP, 2000). Similarly, improvements can be made to increase the efficiency of end-use technology to provide the same level of service while consuming less energy, which effectively reduces energy demand. Another way to achieve energy demand reduction is to enact legislation

requiring or incentivizing energy reduction. Either by improving efficiency or reducing demand, efforts need to be focused on improving the energy system. "Currently trends in energy supply and use are unsustainable – economically, environmentally and socially" (IEA, 2011). The strategy proposed in this research, when implemented, can reduce facility energy's largest demand by reducing facility heating, ventilation, and air-conditioning (HVAC) energy requirements.

Background

The building sector in the United States accounted for 41% of domestic and 7% of global energy consumption in 2010, with HVAC activities consuming approximately 41.4% of the total facility energy consumption (DOE, 2012). The HVAC system in a facility is the system that demands the greatest amount of energy for operation; therefore, improvements to reduce the energy demand of the HVAC system can provide the greatest benefit for facility energy reduction. Reducing HVAC energy demand is achieved by reducing HVAC requirements and improving system efficiency.

The HVAC system functions to maintain an indoor environment suitable for occupant comfort and health. The temperature and humidity in a space generally determines occupant comfort. How much heat that needs to be provided to a space is based on the heat loss. Cooling is provided to counter heat gains and also serves to remove humidity from supply air. Ventilation, or outdoor air, is provided to keep the air in the space fresh and is usually mixed with already conditioned air to improve system efficiency.

When the ventilation requirement in a space is reduced two main types of energy savings are generated: conditioning energy and parasitic energy. Conditioning air involves heating, cooling, humidifying, and dehumidifying as required by the climate and indoor setpoints. Parasitic energy is the energy required to distribute the conditioned air to the end user. Parasitic energy is mainly comprised of fan and pump energy which accounts for approximately 10% of commercial sector energy use (Westphalen & Koszalinski, 1999). Reducing fan energy consumption is an integral part of improving HVAC efficiency.

Facility ventilation also functions to maintain good indoor air quality (IAQ). IAQ, according to the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), "is defined as acceptable when there are no known contaminants at concentrations determined (by cognizant authorities) to be harmful to building occupants, and when a substantial majority (80% or more) of those persons exposed to the indoor air do not express dissatisfaction with its quality" (ASHRAE, 2007c). This definition can be segmented into two parts: the first part is primarily concerned with occupant health, while the second part is concerned with occupant comfort. Occupant comfort is variable and was not addressed in this research effort; therefore, the conditioning energy was not directly considered. Occupant health is jeopardized when indoor contaminant concentrations rise above established thresholds; when these elevated concentrations persist, sick building syndrome (SBS) and other negative effects can result. Contaminant concentrations can be reduced and maintained below the threshold by adhering to the ventilation rates established in ASHRAE Standard 62 (CDC, 2012, EPA, 1991).

ASHRAE standards, which are developed through consensus as defined by the American National Standards Institute (ANSI), have come to be recognized as the industry Standard. ASHRAE Standard 62.1, Ventilation for Acceptable Indoor Air Quality, was significantly revised in 2004; the most notable change made was the modification to the equation for calculating the ventilation required in the breathing zone. Prior to 2004, the ventilation rate was calculated based on occupancy alone (ASHRAE, 2001). After the release of 62.1-2004, the ventilation rate becomes a function of both occupancy and zone size (ASHRAE, 2004). The 2007 edition includes a discussion of carbon dioxide (CO_2)-based demand controlled ventilation (DCV) as a means of reducing energy consumption while maintaining IAQ (ASHRAE, 2007c). The 2010 update to the standard includes minimal revisions and maintains the previously asserted stance on CO_2-based DCV as an energy saving initiative.

The current ventilation rate calculation takes into account both the building and occupants as sources of contamination; therefore, when the zone is unoccupied, the minimum ventilation rate required is determined by the square footage of the zone. Current practice is to calculate the ventilation rate based on zone square footage and maximum design occupancy. The ventilation rate is then established and does not vary based on actual occupancy because the system is supplying the maximum amount of potentially required ventilation. The DCV strategy goal is to optimize the occupant related ventilation requirement based on real-time occupancy. This goal can be achieved with different strategies to determine the occupancy, to include occupancy schedules, occupancy counters, and contaminant sensors (Murphy & Bradley, 2002). Several different contaminants can be monitored to determine the occupancy in a space. The

most commonly used contaminant is CO_2 because “all humans, given a similar activity level, exhale CO_2 at a predictable rate based on occupant age and activity level” which is based in “well-quantified principles of human physiology” (Schell & Inthout, 2001, p. 1). Therefore, accurately determining real-time occupancy is integral to achieving energy savings.

CO_2-based DCV has existed for many years; Emmerich and Persily (1997) conducted a literature review on CO_2-based DCV to consolidate the results of existing research and identify future research needs. Their effort consolidated field tests and simulations on the applicability of CO_2-based DCV in offices, schools, retail stores, public spaces, and residential facilities. None of the research investigated CO_2-based DCV in laboratories. Their research was performed before the 2004 update to the ventilation rate equation; however, their conclusions are still accurate today. Emmerich and Persily (1997) conclude that CO_2-based DCV is most applicable in situations with variable occupancy, a climate that requires heating or cooling throughout the year, and negligible emissions from non-occupant sources. An updated review by Emmerich and Persily (2001) showed that further research was not investigating the applicability of DCV in additional facility types but rather focused on the control algorithms, sensors, and climate impacts on the previously studied facility types. An addition to the updated review was a table showing the energy-cost savings range for various facilities, which is shown in Table 1. Today, there is an updated ventilation standard and still a lack of research regarding the use of CO_2-based DCV in laboratory facilities and the potential savings.

Table 1. Potential Energy-Cost Savings by Facility Type (Emmerich & Persily, 2001)

Facility Type	Energy-Cost Savings Range
Schools	20% to 40%
Lecture Halls	20% to 50%
Open-plan offices (40% average occupancy)	20% to 30%
Open-plan offices (90% average occupancy)	3% to 5%
Assembly halls, theatres, cinemas	20% to 60%

Laboratories have unique HVAC requirements due to the work being performed and the equipment being used in the space. The work being performed often precludes the recirculation of laboratory air and without recirculation there is a greater demand on the supply air to replace the exhausted air. Additionally, the fume hoods used in laboratories to capture contaminants exhaust significant amounts of conditioned air that must be replaced. The previous conditions have led to high minimum air changes per hour (ACH) rates. The United States (U.S.) Occupational Safety and Health Administration (OSHA) asserts that 4-12 ACH is “normally adequate general ventilation” (Phoenix Controls Corporation, 2007, p. 9), while the National Research Council (NRC) states that 6-12 ACH is “normally adequate” (National Research Council, 1995, p. 192). The National Institute of Health establishes 6 ACH as the laboratory minimum and ASHRAE does not prescribe a minimum but states that 6-10 ACH is a general range. The American Conference of Governmental Industrial Hygienists (ACGIH) asserts that ventilation should be based on the contaminant and its generation rate as opposed to ACH (Phoenix Controls Corporation, 2007). In summary, there is no consensus on laboratory ventilation requirements.

Energy savings are achieved when the DCV system reduces the ventilation rate. Laboratories often maintain a high ventilation rate which provides the opportunity to reduce HVAC energy demand. In a laboratory facility, a DCV system can be employed to monitor the laboratory and enable the reduction of ACH while maintaining IAQ; however, there is a dearth of research into the application of DCV in a laboratory setting to achieve energy savings.

Problem Statement

Current practice is to ventilate for the designed maximum occupancy which over-ventilates the space and wastes energy. Laboratories, additionally, have high ventilation demands because fume hoods may be required, depending on the nature of the work being performed. The purpose of this research was to test whether a CO_2-based DCV ventilation strategy can reduce the energy demand of facility HVAC systems while maintaining the recommended IAQ.

Research Questions

The goal of this research was to show how a CO_2-based DCV system can be used as a means to reduce energy demand in laboratory facilities. To address this goal the following primary research question was developed: How can a life-cycle cost effective CO_2-based DCV ventilation strategy be used to reduce energy demand for a laboratory facility when compared to current ventilation practices while maintaining the recommended IAQ? To help answer this question, several investigative research questions were developed. These investigative research questions are listed below.

1. How much energy is reduced as a result of implementing a CO_2-based DCV ventilation strategy?

2. How is IAQ affected when using the CO_2-based DCV ventilation strategy?

3. How much cost savings are generated annually from the CO_2-based DCV ventilation strategy?

These questions were addressed by executing the methodology stated below and explained in detail in Chapter III.

Methodology

This research effort followed a three-phased approach. Phase I analyzed the data generated by the installed DCV system at Wright State University (WSU) to determine the frequency, intensity, and duration of HVAC events during the research period. HVAC events are defined to be anything requiring the HVAC system to increase the amount of ventilation supplied to a zone by greater than 50 cubic feet per minute (cfm). The analysis in phase I yielded a weekly average frequency, intensity, and duration for HVAC events to be incorporated into the model developed in phase II. Phase II included additional data collection and analysis of facility data which enabled the calculation of the fan and overall HVAC energy demand. In phase III, the status quo and DCV demand were compared to determine energy and cost savings. Finally, an economic analysis was performed using Building Life Cycle Cost 5 (BLCC5) software to determine economic feasibility and life-cycle cost effectiveness.

Assumptions/Limitations

Throughout this research effort, there were assumptions made by the researcher and inherent limitations that must be addressed. There were three primary assumptions regarding this research. First, it is assumed that there were no contaminants in the space that could not be identified by the installed sensors. During system installation the

possible contaminants should be identified and monitored appropriately. Second, the dedicated outdoor air system (DOAS) meets the ventilation requirements of the space without any input from the parallel system. Third, the DOAS or parallel HVAC system is capable of meeting the desired laboratory setpoints and maintaining those setpoints within the control limits.

In addition to these assumptions, there were two primary limitations affecting the research. First, the analysis performed is only applicable to laboratories being supplied by a DOAS in parallel with another HVAC system. Second, this analysis is not location specific and cannot, therefore, be used as the only justification for installing a DCV system. These limitations should be considered while before applying using the results of the research.

Organization

The following chapters explore the applicability of using a CO_2-based DCV system to reduce energy consumed by laboratory HVAC equipment. Chapter II discusses the pertinent standards governing ventilation requirements, HVAC system types, and case studies showing how CO_2-based DCV systems have been employed in various facility types. In Chapter III, the methodology for the research is explained for each of the three phases. Chapter IV details the results, while Chapter V concludes this effort with a discussion on the impact of the results achieved. A list of all acronyms and unit abbreviations used is provided in Appendix A as a quick reference for the reader.

II. Literature Review

This chapter expands on the previous section to provide a solid foundation for the research. First, the purpose of the American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) Standard 62 is briefly explained and a comparison is made between ASHRAE Standard 62 and the updated ASHRAE standard 62.1. Specifically, the effects on indoor air quality (IAQ) and the applicability of demand controlled ventilation (DCV) strategies will be explained. Following is a discussion on Heating, Ventilation, and Air-Conditioning (HVAC) system configurations, laboratory specific equipment, and the effects on possible DCV strategies. The chapter concludes with a review of case studies where DCV systems have been applied to different facility types.

Purpose of the ASHRAE Standard 62 Series

The purpose of the ASHRAE Standard 62 series is to establish minimum ventilation rates and other practices to provide an acceptable IAQ (ASHRAE, 2007c). IAQ is based on the occupant's perception of the IAQ and known contaminant concentrations (ASHRAE, 2007a). An occupant's perception of the IAQ in a space is affected by many variables to include light, temperature stratification, noise, air flow, and temperature, which are not a primary concern for maintaining good IAQ and are outside of the scope of this research. Contaminant concentrations are directly addressed through the established ventilation rates to dilute and remove the contaminant from the space. In the user's manual accompanying the release of ASHRAE Standard 62.1-2007, there is an appendix devoted to the implementation and use of a carbon dioxide (CO_2)-based DCV

strategy as an energy conservation measure that modulates outdoor air ventilation rates while maintaining IAQ (ASHRAE, 2007a).

ASHRAE 62.1-2004 Ventilation Rate Changes

Prior to ASHRAE Standard 62.1-2004, ventilation rates were based solely on occupant density; therefore, if there were no occupants in a space, it was acceptable to not ventilate that space. Without any ventilation, however, building-related contaminants would accumulate in a space and reduce the IAQ below acceptable levels. In ASHRAE Standard 62.1-2004, the ventilation rate equation accounts for the additive nature of contaminants and calculations are based on the two primary sources of indoor contaminants: occupants and the building (Stanke, 2006). Equation 1 is the current governing equation for calculating the outdoor air, in terms of cubic feet per minute (cfm), required in the breathing zone (ASHRAE, 2004; ASHRAE, 2007; ASHRAE, 2010 b). V_{bz} is the amount of outdoor air required in the breathing zone in cfm. The first term is the occupant portion of the equation where R_p is the required outdoor flow rate in cfm per person and P_z is the zone population. The second term in the equation is the building related ventilation requirement where R_a is the required outdoor flow rate in cfm per square foot (sq ft) and A_z is the square footage of the zone in sq ft.

$$V_{bz} = R_p P_z + R_a A_z \quad (1)$$

The inclusion of building-related contaminants in the ventilation rate calculation establishes a minimum required ventilation rate proportional to the square footage of the zone. For spaces with low occupant density, the building portion of the ventilation

calculation dominates; conversely, for spaces with high occupant density, the occupant portion will dominate the ventilation calculation. This ventilation rate calculation will vary based on how the supply air is distributed and whether or not the system is heating or cooling. Additionally, this 2004 change was accompanied by a reduction of outdoor airflow rate requirements for certain facility types. The ventilation requirements were reduced as a result of the change in the minimum ventilation rate equation. These reductions also better relate the facility type to the minimum ventilation required while considering energy consumption of the HVAC system. The overall result of these changes, shown in Table 2, is that most occupancy categories have reduced ventilation requirements (Stanke, 2006).

Table 2. Comparison of ASHRAE Standards for Selected Occupancy Categories

	Required Ventilation, cfm/1,000 ft^2	
Occupancy Category	62-1989 through 2001	62.1-2004
Conference/Meeting	1,000	310
Corridors	50	60
Office Space	100	85
Science Laboratories	500	430

Rackes and Waring (2013) studied the impact of using these reduced ventilation requirements with a DCV system on IAQ. They determined that, except for the worst-case buildings, offices implementing a DCV system will not experience significant changes in IAQ. The reduced ventilation requirements and consideration for building-generated contaminants in 62.1-2004 do not adversely affect IAQ; however, these

changes reduce the potential for greater energy savings achieved by a DCV system because the potential ventilation reduction achieved by a DCV system is reduced.

Indoor Air Quality and CO_2

Significant efforts are being made to reduce the energy consumption of operating facility equipment to reduce the environmental impact and overall costs of owning a facility. However, these savings cannot come at the expense of a poor or even hazardous work environment, which will ultimately have more significant costs. There are three main methods for modulating the ventilation to a space to maintain IAQ while reducing energy costs: occupancy schedules, occupancy sensors, and CO_2 sensors (Murphy & Bradley, 2002). An occupancy schedule is implemented by determining the occupancy density for a given time of day and then programming the HVAC system to vary the ventilation based on the pre-determined occupancy. This method is most applicable when the occupants in a facility are on well-defined schedules not conducive to any deviations. Occupancy sensors seek to determine the presence or count the number of occupants in a space. Motion detectors are often used to determine the presence of occupants in a space and return an occupied or unoccupied room status to the DCV system. Counters placed on entry and exit points are generally used occupancy sensors seeking to count the number of occupants in a space. These sensors deliver a real-time occupancy of the room to the DCV system.

CO_2 is an occupant-based contaminant which, if not ventilated sufficiently, can accumulate to concentrations that cause occupants to feel drowsy and lethargic (Mahyuddin & Awbi, 2010). While CO_2 can build to hazardous levels, it is not a primary health concern. Therefore, sensors monitoring CO_2 concentrations are used by DCV

systems to track occupancy and the resulting occupant-based contaminants. Additionally, research supports using CO_2 as an indicator for overall IAQ (Asmi, Putra, & Rahman, 2012; Mahyuddin & Awbi, 2010; Clements-Croome, et al., 2008). However, ASHRAE disagrees with the research to use CO_2 as an indicator of overall IAQ because CO_2 production is not an indicator for volatile organic compounds (VOCs) or other indoor contaminants resulting from building materials and furnishings (ASHRAE, 2010b). Thus, if a space has a strong source of CO_2 that is not occupant based or if there is a significant contaminant, then CO_2 should not be used as the sole indicator for overall IAQ. Yet, ASHRAE does assert that CO_2 is a good indicator for occupant acceptance of the indoor environment because CO_2 production is proportional to bioeffluent production. Furthermore, ASHRAE states that maintaining a CO_2 concentration no greater than 700 ppm above ambient concentrations will provide an indoor environment that satisfies about 80% of visitors to that space (ASHRAE, 2010d).

HVAC Air Handling Systems

HVAC air handling units (AHUs) are used to move air throughout a facility to meet comfort and ventilation requirements. There are three main system types supported by an AHU: single zone, multiple zone, and dedicated outdoor air systems (DOAS). Ventilation zones are determined based on the occupancy category, occupancy density, air distribution effectiveness, and primary airflow per unit floor area (ASHRAE, 2010d). If occupiable spaces have similar requirements for each of these characteristics, they can be classified as a zone because the spaces place equivalent demands on the AHU. A DOAS can be either single or multiple zone system that provides 100% outdoor air and does not recirculate any of the previously conditioned air.

The following seven sections, motors, drives, and fans, total pressure, single zone, multiple zone, DOAS, implications for DCV, and fume hoods, discuss integral impacts to the overall HVAC system and specifically the fan operation. First is a discussion of fan operation and the laws governing its operation. The total pressure in a system drives fan selection and is used to calculate the fan energy while the system configuration changes how the outdoor air rate is determined. The implications for a DCV system of the system configuration are then discussed. Lastly, fume hoods are special equipment found in laboratories which can significantly impact the air requirements in a laboratory.

Motors, Drives, and Fans

Motors in an HVAC system are used to drive the shaft which drives the fan. Motors are sized to be most efficient at the maximum design load or full flow. When a motor operates at a flow other than the design flow, the efficiency of the motor operation changes (Maxwell, 2005). The cube fan law, given in Equation 2, relates flow to power consumption at constant air density. This equation shows how the demand placed on the motor changes with changes in flow. H_1 is the power consumption at the design flow, Q_1, and H_2 is the power consumption at the new flow, Q_2 (Mleziva, 2010). DCV functions to vary the ventilation flow rate based on the occupancy of the space which will change the efficiency of the motor.

$$\frac{H_2}{H_1} = \left(\frac{Q_2}{Q_1}\right)^3 \tag{2}$$

Drives, similar to motors, are sized for, and most efficient at, full flow (Maxwell, 2005). Variable frequency drives (VFDs) are a technology that reduces the power required as the flow is reduced, but does not operate below 20% of the maximum flow rate (Prachyl, 2010). This technology reduces the impact of varying the flow but drive efficiencies still change according to changes in the flow rate.

Fans are used in all-air HVAC systems to distribute air, as required, to meet the requirements in a space. The power consumed by the fan is directly proportional to the volumetric flow (Q) and system pressure (P), shown in Equation 3 (Mleziva, 2010). The density of the air can affect the power consumption of the fan as shown by the second term. “The fan efficiency varies with ventilation flow divided by fan speed” (Mysen, Rydock, & Tjelflaat, 2003). Another fan law, given in Equation 4, shows that flow is proportional to fan speed; therefore, fan efficiency remains approximately constant with changes in flow (Mleziva, 2010). Q_i is the volumetric flow for the corresponding fan speed N_i.

$$AHP = \frac{Q * P}{6356} * \frac{\rho_{air}}{\rho_{standard\ air}} \tag{3}$$

$$\frac{Q_2}{Q_1} = \frac{N_2}{N_1} \tag{4}$$

Total Pressure

Total pressure in HVAC systems is a measure of the resistance in the system that the fan must overcome to move air to its destination and is comprised of velocity and static pressure (Brendel, 2010). It is measured in inches water gauge (in. w.g.) and varies significantly based on how the HVAC system is designed. Total pressure exists on the supply and exhaust sides of the HVAC system where the exhaust side total pressure is usually 0.5 in. w.g. less than the supply side (Aircuity, 2012). Understanding the total pressure in the system is paramount to reducing energy consumption because the energy required to move air is determined based on the total pressure in the system and the flow of the air through the system. A greater total pressure requires more energy to move the air to its destination because there is a greater resistance.

The last fan law shows how a change in static pressure, P_{si}, is proportional to the ratio of change in flow, Q_i, squared and is given in Equation 5 (Mleziva, 2010). DCV modulates the flow in a system to meet the real-time requirements in a space which means that the static pressure in a system will also change. This relationship is important because fan power consumption is directly proportional to both flow and total pressure as shown previously in Equation 3. Therefore, as the DCV system modulates flow the static pressure in the system will change which directly affects the power consumption of the fan.

$$\frac{P_{s2}}{P_{s1}} = \left(\frac{Q_2}{Q_1}\right)^2 \qquad (5)$$

Single Zone

A single zone HVAC system is the simplest to design and operate because there is one AHU providing air to a space with homogenous loading. The overall system outdoor air intake rate (V_{ot}) is equal to the zone outdoor air rate (V_{oz}). The zone outdoor air rate is related to the breathing zone air rate (V_{bz}) based on the zone air distribution effectiveness factor (E_z) as shown in Equation 6 (ASHRAE, 2010d).

$$V_{oz} = \frac{V_{bz}}{E_z} \tag{6}$$

Because of these relations, after accounting for air distribution effectiveness, the ventilation requirement for a single zone system is based solely on the zone floor area and the zone population. Therefore, modulating the outdoor air intake based on the real-time zone population provides the opportunity for energy savings in a single zone system while still meeting codified ventilation requirements. All systems require ventilation controls and single zone systems typically utilize constant air volume (CAV) control measures, as opposed to variable air volume (VAV) controls, which provide air at a constant volume. Thus, each space within the zone receives conditioned air at the same volumetric rate.

Multiple Zone

Multiple zone systems exist when a single AHU provides air to multiple zones simultaneously. These zones have differing ventilation demands which require varying outdoor air rates. Because of this variability, multiple zone systems commonly utilize

VAV boxes as terminal control units in each space to modulate the supply air into that space. While VAV boxes provide variable supply air to a space as required, there is an inherent inefficiency in the system because multiple zones with different ventilation requirements are being supplied by a single AHU with fixed ventilation. This inefficiency becomes apparent in the process to determine the required outdoor air flow rate.

The first step to determine the required outdoor air flow rate is to determine system efficiency (E_v). The system efficiency is based on the maximum primary outdoor air fraction (Z_{pz}) which is calculated by determining the outdoor airflow (V_{oz}) for each zone divided by the primary airflow for each zone (V_{pz}) (ASHRAE, 2010d). The system efficiency is based on the zone within the system with the greatest demand. Therefore, the percentage of outdoor air supplied is going to be greater than the percentage of outdoor air required for each zone not placing the greatest demand on the system. The uncorrected outdoor air rate (V_{ou}) is then calculated as shown in Equation 7 (ASHRAE, 2010d). The first summation calculates the occupant-based ventilation requirement where R_p is the required outdoor flow rate in cfm per person and P_z is the zone population. The second summation in the equation is the facility-based ventilation requirement where R_a is the required outdoor flow rate in cfm per square foot (sq ft) and A_z is the square footage of the zone in sq ft.

$$V_{ou} = D \sum_{all\ zones} R_p P_z + \sum_{all\ zones} R_a A_z \tag{7}$$

The variable *D* in Equation 7 accounts for occupant diversity. Occupant diversity considers that each space is not at its design occupancy rate simultaneously. Lastly, the system outdoor air rate (V_{ot}) is calculated by dividing the uncorrected outdoor air rate (V_{ou}) by the system efficiency (E_v) (ASHRAE, 2010d). Because ventilation demand calculations for multiple zone systems, like single zone systems, establish breathing zone ventilation rates based on the greatest design demand, energy is wasted due to periods of over-ventilation. Additionally, the possible ventilation demand variation between zones in multiple zone systems increases the inefficiency of the system.

Dedicated Outdoor Air Systems

Dr. Stanley Mumma is one of the pioneers for and a leading proponent of DOAS. His work has helped to establish how DOAS can be implemented with parallel systems to meet thermal and ventilation requirements in a space. Furthermore, a 2002 paper discusses how a VAV system with DCV compares with a DOAS to meet the ventilation requirements of a space. Mumma (2002) shows that a DOAS unit is able to meet ventilation requirements much more efficiently than a VAV system and that a DCV system does not significantly improve DOAS efficiency. However, this analysis does not consider the unique ventilation requirements of a laboratory. Specifically, laboratories require significant amounts of outdoor air without recirculation. These requirements increase the energy saving potential of a DCV system.

As previously stated, DOAS can be either single or multiple zone systems that supply only outdoor air to a given zone or zones. Ventilation requirements for these systems are calculated similar to those in a single zone system with the exception that the

system outdoor air intake (V_{ot}) is equal to the sum of zone outdoor air flow rates (V_{oz}) as shown in Equation 8 (ASHRAE, 2010d).

$$V_{ot} = \sum_{all\ zones} V_{oz} \tag{8}$$

A DOAS provides the required air to fulfill ventilation requirements, while, typically, separate units condition the air to meet the thermal requirements of the space (Stanke, 2004). The outdoor air provided by the DOAS is thermally neutral and the supply air (return air and outdoor air) provided by the parallel HVAC system is conditioned to meet the thermal requirements of the space. Design guidance in the Unified Facilities Criteria (UFC) 3-410-1 outlines a preferred moisture control method that splits the ventilation and cooling requirements using a DOAS unit for any zone requiring greater than 1000 cfm of ventilation. Additionally, UFC 3-410-01 specifics that the DOAS unit should be sized to handle the latent loads when cost effective. UFC documents are used throughout the Department of Defense (DOD) and provide "planning, design, construction, sustainment, restoration, and modernization criteria" (WBDG, 2014). For this method, the DOAS unit is designed to specifically handle ventilation and humidity requirements in the zone while the AHU satisfies the thermal requirements of the space, specifically the cooling and sensible heat loads. This division of work can reduce the energy demands of the system depending on the climate.

Figure 1 shows how DOAS is typically integrated with a parallel VAV system. However, there is not a 100% consensus on how the two systems should combine to

supply air to the space (Greenheck, 2007; Mumma, 2014). This system setup is similar to the setup described in Mumma (2001). Yet, as described in Figure 1 the DOAS is solely responsible for ventilation while the parallel system is meeting both the sensible and latent loads. This research is not location specific therefore it is unknown if decoupling the latent load is cost effective, as required by UFC 3-410-01. The DOAS output can also be combined with the parallel HVAC output before being supplied to the space to potentially reduce equipment costs and to better control room air distribution.

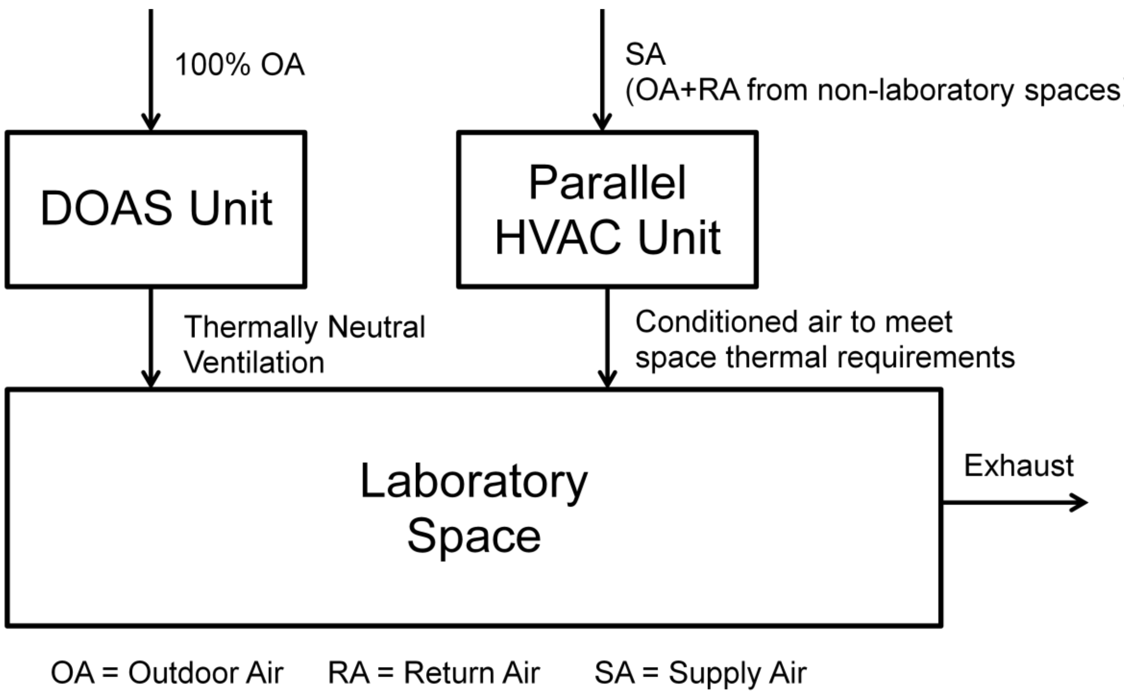

Figure 1. DOAS and Parallel System Setup

According to ASHRAE Standard 90.1, *Energy Standard for Buildings Except Low-Rise Residential Buildings*, energy recovery is required based on the climate zone and the design supply fan airflow rate; however, laboratories have restrictions on how the

exhaust air can be used in energy recovery (ASHRAE, 2010a). This analysis does not consider an energy recovery system and the resulting pressure drop because the analysis is not location specific. If an energy recovery system is required based on a specific climate zone and design supply fan airflow rate, then the pressure drop from including the system should be accounted for throughout the analysis.

Implications for DCV

The thermal and ventilation demands of a facility primarily determine the type of system that meets the requirements most economically. Installing a DCV system can reduce the cost of HVAC system operation; however, there are many factors that affect the economic feasibility of the DCV system. First among these factors is the system itself.

Single zone systems employing CAV controls provide the greatest opportunity for savings utilizing tested and proven technology. The CAV control ensures that the zone is not under-ventilated, but it results in the greatest amount of energy waste by ventilating the zone based on maximum design occupancy when in operation. DCV can be used in this situation to determine the actual occupancy of the zone and provide the minimal outdoor air required, which reduces the demand on the conditioning and distribution systems.

Multiple zone systems using VAV controls can be more efficient than CAV single zone systems; yet, multiple zone systems are inherently inefficient because the different zones may require different outdoor air requirements. DCV can be used to reduce this inefficiency by modulating the outdoor air demand of the most critical zone, thereby reducing the overall conditioning requirements of the system and the excess in the non-

critical zones. ASHRAE is sponsoring research to determine "if and under what conditions CO_2 DCV can be efficiently and effectively implemented with multiple zone systems" (ASHRAE, 2010d). However, research has tested and ASHRAE has approved a procedure to dynamically reset outdoor air rates for multiple zone systems without using CO_2-based DCV (ASHRAE, 2010d). Additionally, many research efforts into the implementation of a multiple zone DCV system have been undertaken without a clear best solution (Liu, Zhang, & Dasu, 2012).

A DOAS takes on the inefficiencies of whichever configuration is being employed; furthermore, energy demand is increased because all of the outside air must be conditioned without recirculation. Yet, a DOAS might be required due to the specific ventilation requirements of the space. In this system, the energy required to provide a single unit of conditioned air to the space is greater than recirculating systems; therefore, the impact of ventilation reduction due to a DCV system will also be greater.

Recall that the parasitic energy in an HVAC system is the energy required to power the pumps and fans. This energy represents 20% to 60% of the total HVAC electrical energy demand (Westphalen & Koszalinski, 1999). A DCV system aims to reduced both parasitic and conditioning energy; however, the focus of this effort is the DCV impact on the reduction in parasitic energy consumption, specifically, fan energy.

Fume Hoods

A laboratory setting requires that fume hoods also be considered when determining ventilation in the space. Fume hoods function to contain and exhaust airborne contaminants or gases from the facility. During use, an individual opens the sash to perform the functions required within the hood. The air flow into the fume hood

at the opening is considered the face velocity and can range from 60 to 150 feet per minute (fpm) depending on which standard is followed (Phoenix Controls Corporation, 2007). Energy costs of fume hood operation are directly proportional to face velocity, and a higher face velocity does not necessarily translate to greater containment because of turbulence created by the worker (National Research Council, 1995). Therefore, the face velocity of the hood must be determined while balancing laboratory safety with energy costs of operation. Additionally, all of the air exhausted by the hood must be replaced with conditioned air which increases demand on the HVAC system and operating costs.

Fume hoods change how ventilation is determined in laboratories when compared to other spaces. According to the ASHRAE Applications Handbook, minimum ventilation rates requirements are considered third in a laboratory. The total amount of air exhausted is considered first followed by any thermal requirements for internal heat gains (ASHRAE, 2007b). However, there is no requirement to leave the fume hood on when it is not in active use or being used to store hazardous substances; thus, under this condition, the thermal requirement dominates. However, while meeting thermal requirements, the ventilation rate can be reduced to the minimum requirement. Therefore, a CO_2-based DCV system can be used to control laboratory ventilation provided the system meets the ventilation requirements when the fume hood is in operation.

Case Studies for Different Facility Types

The climate and type of facility employing the DCV system have a significant impact on the effectiveness of the system. Emmerich and Persily (2001) assert that DCV

implementation and use is most effective for facilities with highly variable and unpredictable occupancy schedules, minimal contaminant emissions from non-occupant sources, and in climates that require constant heating or cooling. The following case studies show that CO_2-based DCV systems have been implemented in various facility types to achieve energy savings.

Gymnasium

The first case study examines a CO_2-based DCV system in an elementary school gymnasium located in West Lafayette, Indiana, which has a humid continental climate (Ng et al., 2011). The gymnasium ventilation system was a single zone CAV system and was selected for study because the highly variable occupancy in the space provided the potential for energy savings when utilizing a DCV strategy in place of a fixed ventilation strategy. The existing fixed ventilation strategy operated with a 50% open outdoor air damper which over-ventilated the gymnasium (Ng et al., 2011).

Their experiment was conducted for 42 days during July and August of 2010. Predictive models were developed based on data collected on 17 August 2010. On this day, the high temperature reached 82°F while the low was 59°F.   Measurements for temperature, CO_2, and relative humidity were taken from wireless wall-mounted sensors. The CO_2 sensor was laboratory calibrated to an accuracy of ±30 parts per million (ppm) (Ng et al., 2011). Figure 2 shows the location of the sensors and the layout of the gymnasium. As shown, two CO_2 sensors, identified by a circle around “CO2,” were located in the gymnasium within the breathing zone. A third sensor was located in the main return air duct for comparison with the wall-mounted sensors. Wall-mounted sensors readings can be artificially increased if an individual breathes heavily in close

proximity of the sensor. This close proximity prevents the exhaled CO_2 from mixing with the ambient air which results in an artificially high CO_2 measurement. In this experiment, the researchers assumed that the supply air was distributed with 100% effectiveness. The computer recording the data is identified by a square around "PC" and the base station is identified by a circle around "BS."

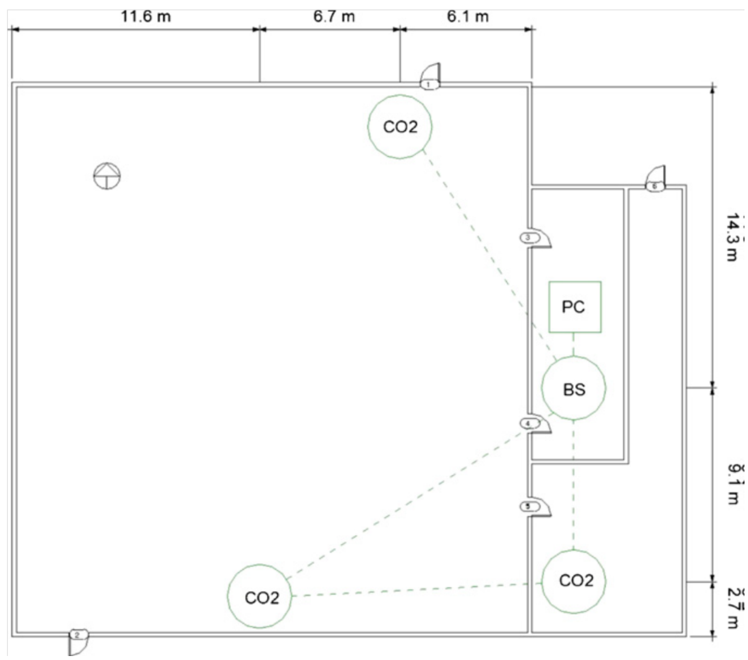

Figure 2. Gymnasium Floor Plan and Sensor Layout (Ng et al., 2011)

Occupancy was counted on selected days to verify occupancy detection calculations. The researchers considered two occupancy prediction approaches: steady-state and transient. The steady-state algorithm assumes that a steady-state CO_2 differential between the space and the outside air has been reached. The researchers determined that the steady-state equation produced a lag time of approximately 30 to 40 minutes when responding to a change in occupancy. Additionally, because multiple

hours are usually required to reach 90% of the steady-state, the algorithm routinely underestimates actual occupancy (Ng et al., 2011). The transient algorithm is the mass balance of CO_2 at the AHU discretized. This algorithm was determined to be highly responsive to changes, yet less precise with a tendency to overestimate occupancy (Ng et al., 2011). The occupancy profile, shown in Figure 3, compared the two different occupancy prediction approaches considered by the researchers with the actual occupancy on 17 August 2010. The lag and underestimation of the steady-state algorithm compared with the oscillatory nature and overestimation of the transient algorithm can be clearly discerned. Because of the increased accuracy and responsiveness of the transient algorithm, the researchers utilized its model to determine ventilation rates and energy consumption.

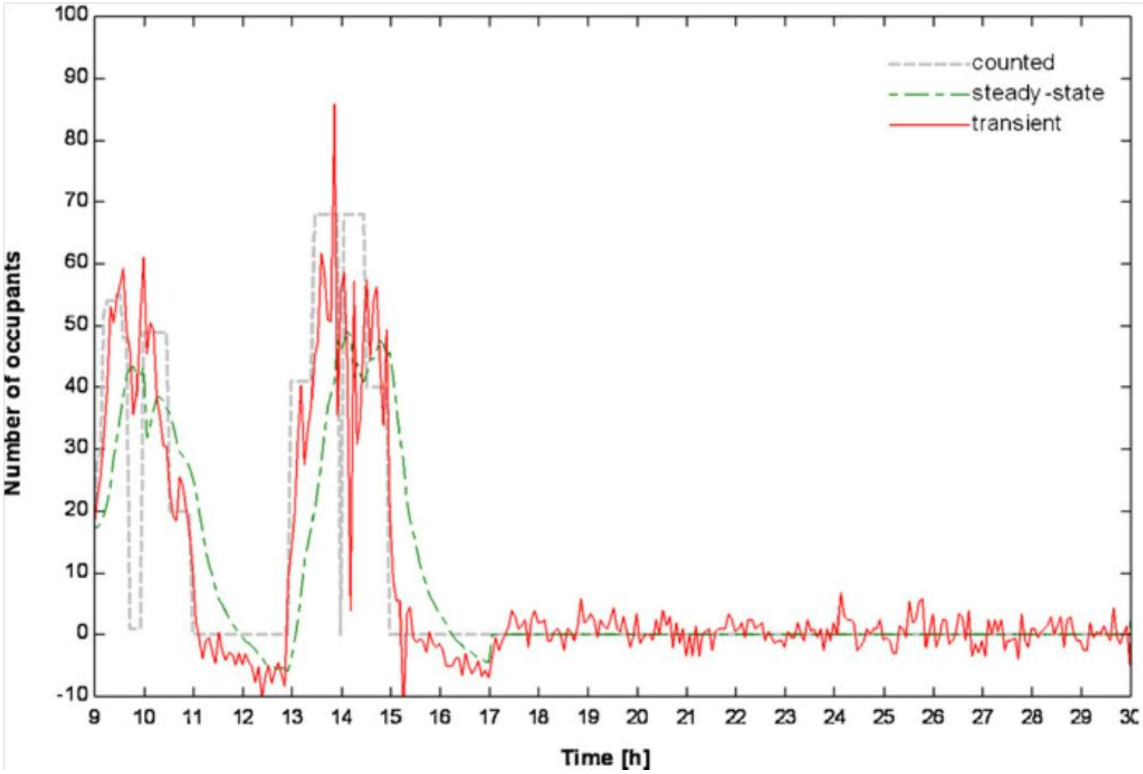

Figure 3. Counted Versus Predicted Occupancy Profile (Ng, et al., 2011)

The baseline energy conservation ventilation strategy was a fixed ventilation rate of 5% outdoor air (Ng et al., 2011). This strategy disregarded all standards and Figure 4 shows that the strategy does not provide sufficient ventilation during peak hours. The ASHRAE standard 62.1 proportional strategy is recommended and will always meet the minimum required ventilation rate; however, the lethargic nature of this strategy leads to substantial periods of over-ventilation (Ng et al., 2011). The final two strategies both use the transient algorithm to determine real time occupancy; yet, one strategy uses the revised ASHRAE standard 62.1 while the other uses the outdated ASHAE standard 62. As shown, using the ASHRAE standard 62 allows the system to turn ventilation off when zero occupancy is detected. Furthermore, the higher peaks are a product of the higher per

person ventilation requirements. Conversely, the new ASHRAE standard 62.1 relaxes the per person requirements, thereby resulting in lower peaks but establishing a minimum required ventilation when the space is unoccupied. Each strategy tested was able to maintain the CO_2 concentrations below the ASHRAE standard 62.1 recommended limits.

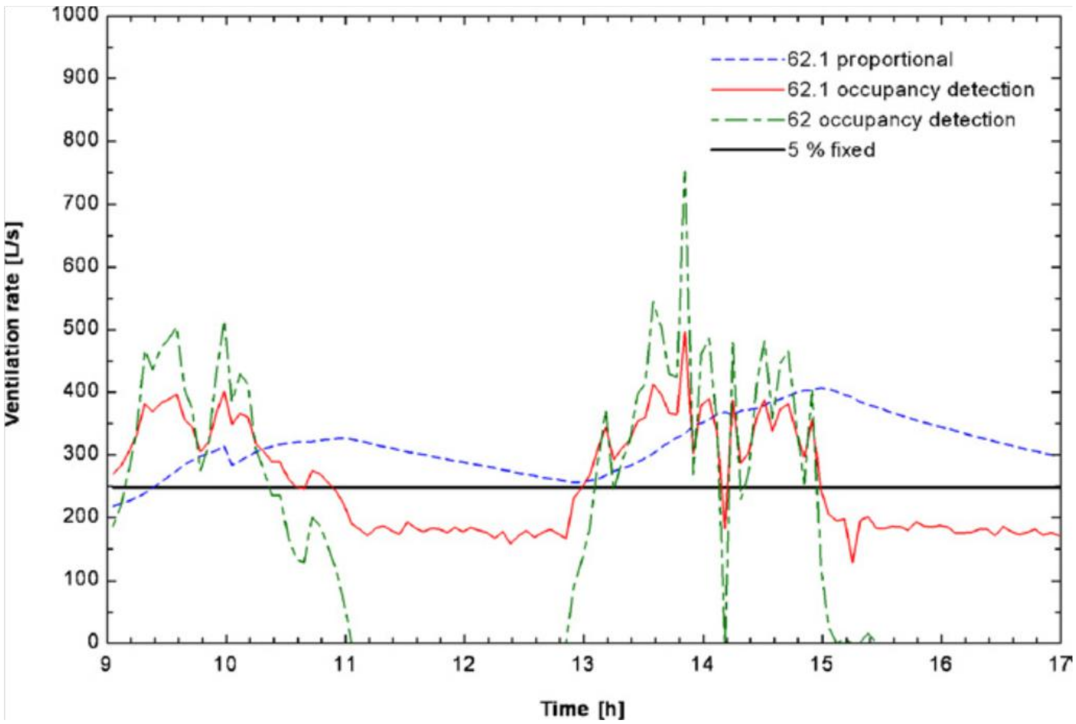

Figure 4. Simulated Ventilation Rates (Ng et al., 2011)

Ng et al. (2011) determined energy consumption based on the cooling coil in the AHU, neglecting fan energy, and determined that the energy reduced using ASHRAE standard 62 yielded savings of 1.86% while ASHRAE standard 62.1 yielded 0.03% savings when compared to the 5% fixed ventilation strategy. These savings are small because the occupancy detection strategies are being compared to another energy saving strategy that does not consider minimum ventilation requirements. Additionally, Ng et

al. (2011) suggest that the coil energy savings could be increased if measurements were taken on a hotter day or in a more severe climate that places greater demand on the cooling coil.

In their experiment, Ng et al. (2011) does not consider the fan energy reduction savings achieved for any of the strategies considered. In this research effort, the primary focus is on the fan energy reduction achieved by a DCV system. The resulting conditioning energy is determined based on the total fan energy. The following section details a study by Nielsen and Drivsholm (2010) in which the fan energy reduction savings are the only savings considered.

Residential

Nielsen and Drivsholm (2010) undertook a study to determine if a simple CO_2-based DCV strategy could be applied in a single-family home to realize energy savings without adversely affecting IAQ. The DCV strategy applied in this study used CO_2 concentrations to determine when the house was occupied and humidity measurements to ensure that IAQ was not reduced because of the lower ventilation rates. When occupied, the ventilation rate was set to 216 cubic meters per hour (m^3/hr), as required by Danish Building code, and when unoccupied the ventilation rate was reduced to 80 m^3/hr (Nielsen & Drivsholm, 2010). This approach to CO_2-based DCV was different from the one employed by Ng et al. (2011), because Nielsen and Drivsholm (2010) were not trying to vary the ventilation rate based on real time occupancy; instead, they were using CO_2-based occupant detection to switch between the unoccupied minimum and the occupied maximum ventilation rates.

The researchers determined that the optimal CO_2 concentration differential between exhaust and outdoor air for determining occupancy was 150 ppm. If an outdoor CO_2 concentration of 400 ppm is assumed, this strategy maintains CO_2 concentrations below the ASHRAE standard 62.1 maximum. Figure 5 shows how CO_2 concentrations accumulate at a constant minimum ventilation rate. This figure shows that the system required less than one hour to determine occupancy when four people enter the home, while it took just under three hours to determine occupancy for one individual. Additionally, the humidity difference between exhaust air and outdoor air was tested and 2 grams per kilogram (g/kg) was determined to be the optimal setting to switch on the high ventilation rate for humidity control (Nielsen & Drivsholm, 2010).

The study implemented the DCV strategy in a 140 square meters (m^2) single family house occupied by two adults and two children where the adults and children were away from the house during the day for work and school, respectively, throughout February, March, and April of 2009 (Nielsen & Drivsholm, 2010). Average temperatures for Denmark in these months range from 0°C to 6°C (Weatherbase.com, 2013). For this experiment, the CO_2 and humidity sensors were located in the exhaust air duct and the outdoor air intake; the ventilation rates were controlled by the speed of the fan (Nielsen & Drivsholm, 2010).

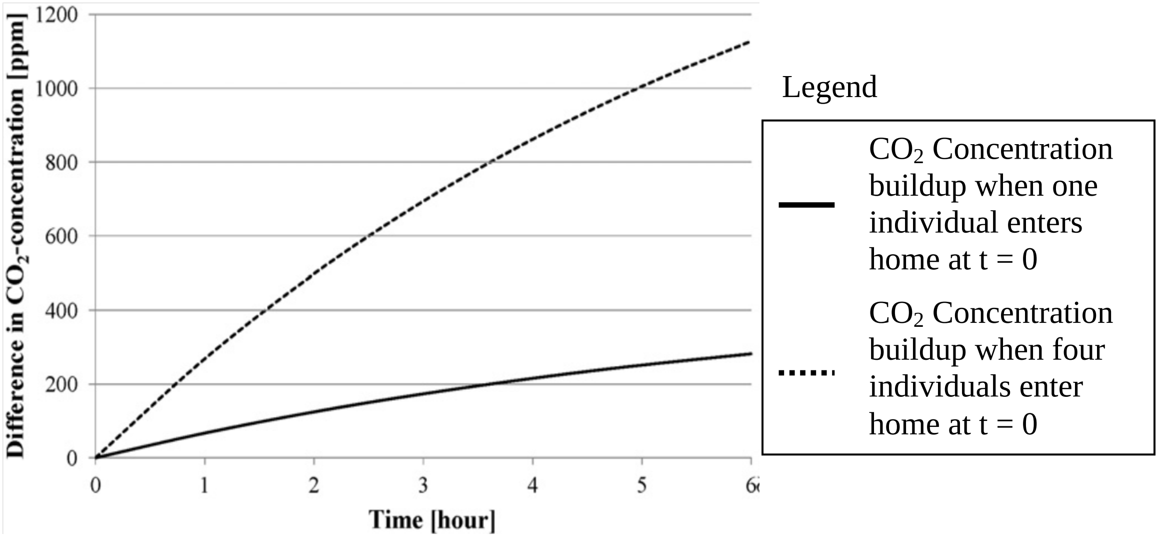

Figure 5. Time Required to Determine Occupancy (Nielsen & Drivsholm, 2010)

Figure 6 shows that when the system used the optimal thresholds of 2 g/kg for humidity control and 150 ppm for occupancy detection, the fan can operate at the lower speed for various periods of time each day (Nielsen & Drivsholm, 2010). When combined with the humidity restraint, Nielsen and Drivsholm determined that the fan can operate at the lower rate 37% of the time without adversely affecting IAQ.

Reducing the fan speed by 136 m^3/hr for more than a third of the time reduced the energy demand of the ventilation by 35% (Nielsen & Drivsholm, 2010). Additional energy reduction could have been reported if the researchers had included the energy savings from the reduction in the amount of air that must be conditioned before it was ventilated to the house. Specifically, to condition one unit of air with a high heating load, requires substantial energy. Yet, this case study shows that an appropriate DCV strategy can be used in a situation with variable occupancy to reduce the energy demand of the HVAC system while maintaining IAQ.

Fan speed reduction savings can be generated regardless of the location because the ventilation requirement is independent of any thermal requirements. Additional savings based on conditioning energy required can vary with location. The research conducted by Nielsen and Drivsholm establish a foundation on which the methodology described in Chapter III is based. Further, this study into laboratory ventilation will not be location specific because the space thermal requirements are not considered.

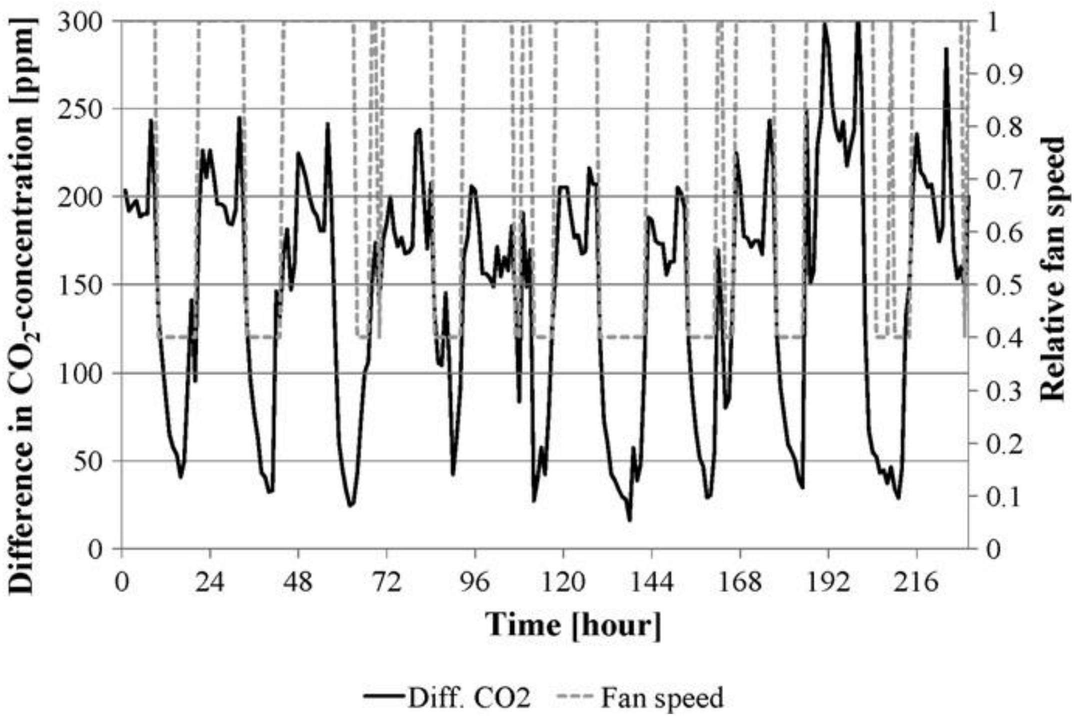

Figure 6. CO_2 Concentration Difference against Fan Speed (Nielsen & Drivsholm, 2010)

School

Schools have a significant amount of occupancy variability due to class schedules and can therefore potentially benefit from implementing a DCV strategy. Mysen, Berntsen, Nafstad, and Schild (2005) researched the potential benefits from implementing several DCV strategies in Norwegian primary schools. Their study investigated the

benefits of a CO_2-based DCV system and an infrared (IR) occupancy sensor based DCV system when compared to the existing CAV strategy. The CAV strategy provided air 10 hours per day based on the design of 30 occupants per room. According to Norwegian code, 7 liters per second (L/s) are provided per person and an additional 1 liter per second per square meter ($L/s/m^2$) is provided for building source contaminants (Mysen et al., 2005). This situation was equivalent to the updated ASHRAE standard 62.1 by accounting for occupant-based and building-related contaminants.

The IR-based DCV strategy is a bimodal strategy which provides the minimum air required for building source contaminants when the space in unoccupied and provides the maximum design calculated airflow when the space is occupied (Mysen et al., 2005). This strategy is much like the strategy used by Nielsen and Drivsholm (2010) in their study of DCV applications in residential homes. The CO_2-based DCV strategy provided the minimum air required until the CO_2 concentration reached 900 ppm. Once at 900 ppm, the system regulated ventilation as required to maintain a concentration of no more than 900 ppm until the concentration dropped below 700 ppm. Once below 700 ppm, the system reset to the minimum required ventilation rate (Mysen et al., 2005).

Their research was performed from 5 March 2002 to 17 June 2002 at 81 randomly selected schools in Oslo, Norway. Average temperatures ranged from 0°C to 16°C for the selected time frame (Weatherbase.com, 2013). After an inspection of 157 classrooms, it was determined by the researchers that the mean classroom occupancy time in a day was four hours and the mean occupancy was 22 individuals (Mysen et al., 2005). These mean values indicate that the classrooms were being over-ventilated by 56 L/s and that the system was running at maximum design occupancy for 6 hours when

the space was unoccupied. Figure 7 shows the comparison of air volume supplied by each of the ventilation strategies, and it is clear that the CAV system provides significantly more air to the space than the other two control strategies. Each cubic meter of air requires energy for conditioning and ventilation, thereby resulting in increased costs. The difference between the line representing CAV and the lines representing the DCV systems is the amount of ventilation reduced. The reduction in ventilation directly relates to energy savings. Figure 8 expands on Figure 7 by calculating the energy required based on air volume. For their study, energy consumption was calculated based on the fan energy and the energy for space heating (Mysen et al., 2005). By considering both fan and heating energy a more accurate representation of the energy savings is achieved, as opposed to only cooling energy, Ng et al. (2011), or only fan energy, Nielsen and Drivsholm (2010). The energy required for the DCV strategies is then compared to the energy required for the CAV baseline. As shown, the use of a DCV strategy can generate substantial savings when compared to a CAV ventilation strategy. Both the CO_2-based DCV system and IR-based DCV system reduce energy use by 38% and 51% of CAV, respectively, for 10 hours of daily operation.

The highly variable occupancy density and patterns of the schools provided an ideal situation for a DCV strategy to achieve energy savings. Classrooms were not being occupied by the number of individuals for which the HVAC system was designed, thus leading to over-ventilated classrooms. Additionally, the classrooms were being occupied for less than half of the designed occupancy time, which further increased the over-ventilation. Furthermore, significant heating loads were not placed on the system throughout the period of study which would have led to even more energy savings.

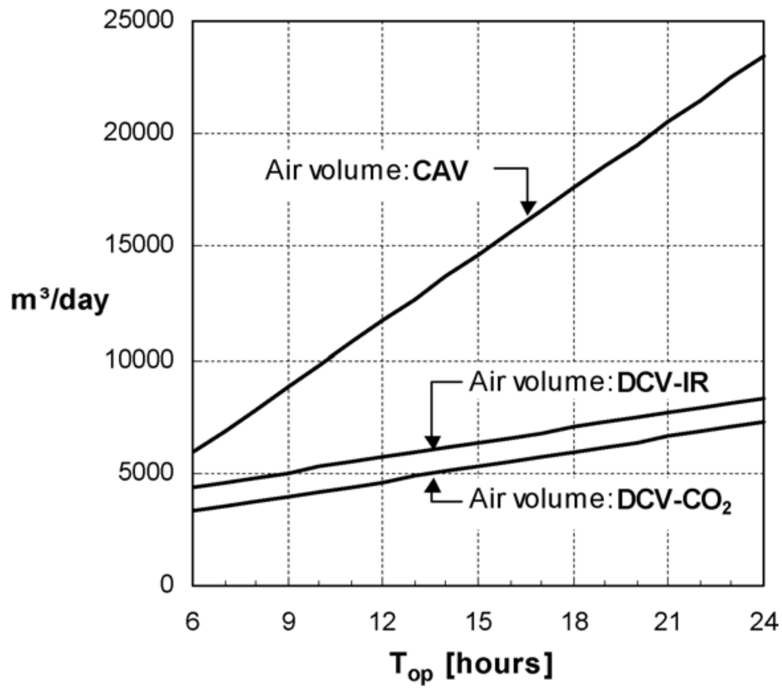

Figure 7. Air Volume Supplied by Different Control Strategies (Mysen et al., 2005)

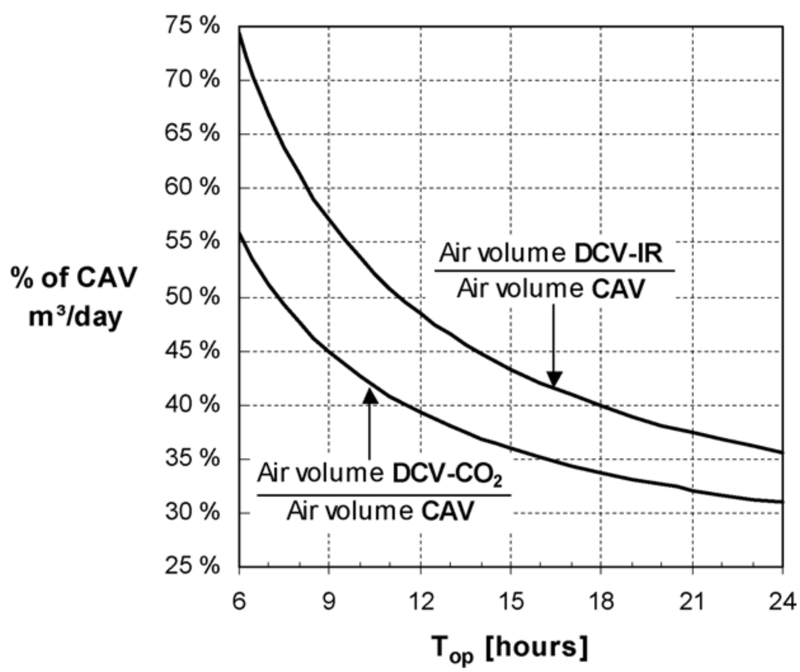

Figure 8. Energy Savings per Year (Mysen et al., 2005)

Laboratory

As previously discussed, there is no consensus on minimum laboratory air change and ventilation rates; however, the accepted practice is to establish a higher air change rate to keep the laboratory continually supplied with fresh ventilation. To support reducing laboratory air change rates, Sharp (2010) reports on a study that collected IAQ data from laboratories utilizing DCV systems. The data was collected from the fall of 2006 until January 2009 on 15 different laboratories located throughout the U.S. The total sample was approximately 1.5 million hours of IAQ laboratory data (Sharp, Demand-based control of lab air change rates, 2010). Sharp (2010) determined that the laboratory air change rate could be reduced approximately 99% of the time. From the reduced baseline, the average laboratory room required increased ventilation 1.5 hours per week to maintain acceptable IAQ. His study shows that laboratory air change rates can be reduced without affecting safety in the laboratory environment. Additionally, reducing laboratory air change rates provides an opportunity to achieve energy savings.

A DCV system was installed in a laboratory at the University of California-Irvine (UCI) as a pilot study. After installation, the average daily airflow was reduced by greater than 30% when using the DCV system compared to the status quo. The reduction in airflow resulted in fan energy reduction of approximately 40% (Bell, Matthew, & Van Greet, 2010).

A best practices guide created by a joint U.S. Environmental Protection Agency and U.S. Department of Energy program asserts that "nearly half of the electrical energy use in a typical laboratory can be attributed to ventilation" (Bell, 2008). The guide further states that DCV can be used to meet real-time ventilation requirements and has

the additional benefit of being able to monitor and detect hazards in the air. A DCV strategy enables HVAC designers to optimize the laboratory ventilation rate to meet safety requirements while increasing energy efficiency; yet, many laboratories have not investigated if this technology is able to help meet mandated energy reduction goals and reduce HVAC operation costs.

Summary

This chapter has established the foundation for implementing a CO_2-based DCV system and discussed how different types of CO_2-based DCV systems were implemented in varying facility types to achieve energy savings without adversely affecting IAQ. A laboratory facility has a 100% outside air requirement which increases the cost per volume of air conditioned and provides an opportunity for energy savings. CO_2-based DCV ventilation strategies have been shown to reduce energy consumption associated with facility HVAC use for different facility types and different control strategies. However, there is a lack of study into the use of CO_2-based DCV systems to reduce energy consumption in HVAC systems supporting laboratory facilities. A laboratory is ideal for implementation of a DCV system because of the large air volume requirement and the use of a dedicated outdoor air system.

III. Methodology

This research effort utilized descriptive statistics to determine heating, ventilating, and air-conditioning (HVAC) energy reduction in laboratory facilities utilizing a carbon dioxide (CO_2)-based demand controlled ventilation (DCV) system. This chapter discusses the three phases of the methodology depicted in Figure 9. In Phase I, evidence is given to support using the Wright State University (WSU) DCV laboratory data as an estimate for laboratories in general. Further, the data is analyzed to determine a typical week of supply air demand for the laboratories. Phase II defines typical characteristics of Air Force laboratories by analyzing the Battlespace Environment Laboratory (BEL) located at Kirtland Air Force Base in New Mexico. Using these typical laboratory characteristics, a range of laboratory configurations are considered and the minimum ventilation baseline is calculated. In phase III, the results of the first two phases are synthesized and used in a life-cycle cost analysis (LCCA) to determine the potential energy savings when using a DCV system in laboratories.

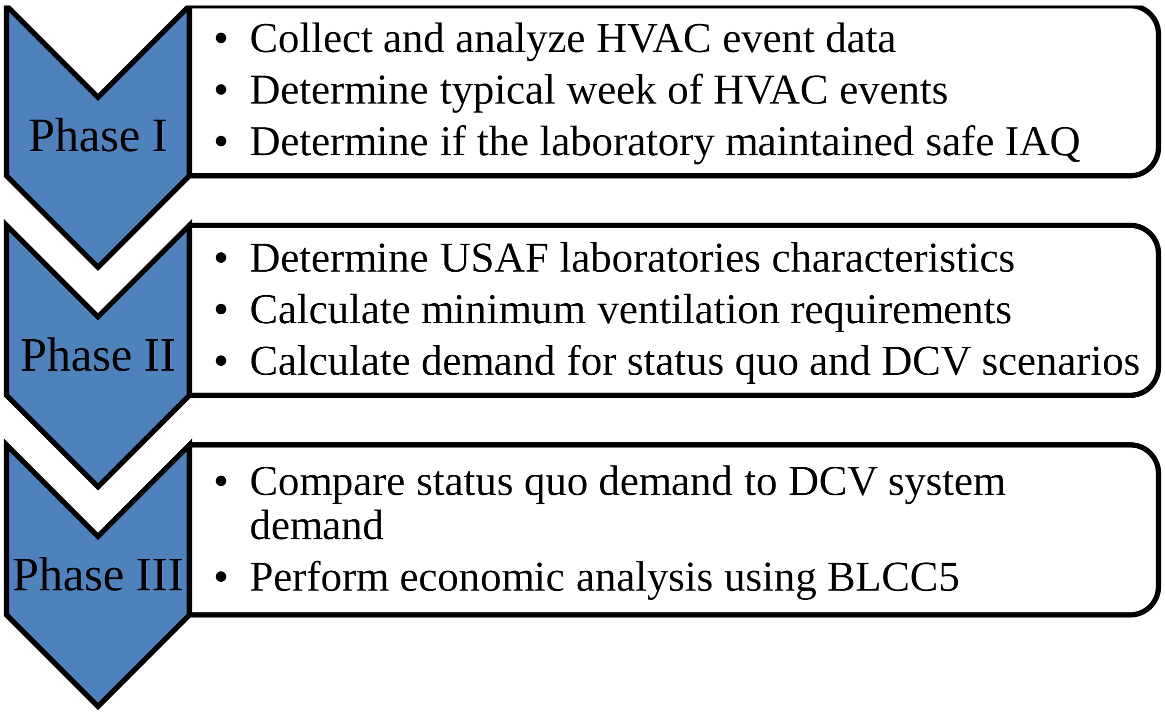

Figure 9. Methodology Process

Phase I – HVAC Events

The first phase of the methodology process seeks to determine the frequency, duration, and intensity of HVAC events. A HVAC event occurs when laboratory conditions change, thus requiring a change in the amount or condition of air supplied to a space. HVAC events are the result of changes in one or more of the following four conditions (ASHRAE, 2007b). Laboratory safety is the first condition and is always considered by the system. The second condition is to maintain room pressurization requirements followed by the thermal comfort of the occupants, the third condition. The last condition is to maintain minimum ventilation as required by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) standard 62.1 (ASHRAE, 2007b). These conditions drive the HVAC events which cause the HVAC

system to deviate from the established baseline. The results from phase I will show if the CO_2-based DCV system is able to maintain safe IAQ in the laboratory.

Data Collection

Wright State University (WSU), located in Dayton, Ohio, installed CO_2-based DCV systems in three laboratories in the summer of 2013: Bio Science I, Diggs Laboratory, and Oelman Hall. The data used in this research was retrieved from the DCV systems monitoring these three laboratories from 30 September to 3 November 2013. The laboratory spaces in each facility are supplied by an air handling unit (AHU) using a fixed 100% outdoor air intake. The AHU functions similar to a DOAS unit by supplying only outdoor air; however, while a DOAS unit typically provides thermally neutral outdoor air to meet ventilation requirements, the AHU is meeting the required ventilation as well as the latent and sensible loads of the laboratory spaces. Table 3 gives general characteristics for each laboratory facility.

Table 3. WSU Laboratory Characteristics

Facility	Average sq ft/zone	Number of Zones	Pre-DCV ACH, Day	Pre-DCV ACH, Night
Bio Science I	862	18	10	6
Diggs Laboratory	1161	10	8.05	4
Oelman Hall	416	12	11.46	4

Once installed, the systems monitored the conditions in the laboratory spaces and modulated the variable air volume (VAV) terminals and venturi valves from the baseline as required to respond to any HVAC event. Venturi valves provide another way besides

VAV terminals to determine airflow in an HVAC system. As discussed in the literature review, there is a hierarchy of needs that the DOAS must satisfy in which safety is always considered first. Safety is monitored specific to the use of the laboratory. The systems at WSU monitor the Total Volatile Organic Compounds (TVOCs) and small particulate matter in the air to maintain a safe laboratory. To ensure that the system can maintain laboratory safety, the installed sensors must be verified to ensure they are able to detect if a hazard is in the air. The second parameter that must be met is room pressurization. Specifically, the total exhausted air (i.e. fume hood and general exhaust) plus the room offset must be supplied to the laboratory to maintain the desired room pressurization. Most laboratories are maintained at a negative pressure so that any contaminants released into the air are contained within the space by the pressurization. The room offset is the magnitude of pressurization that is maintained by the HVAC system and is 100 cfm for this effort.

The thermal comfort of room occupants is the third parameter that must be satisfied and does not necessarily increase the volume of air being supplied to the space. The same volume of air can be cooled or heated to a greater extent to meet the thermal requirements of the occupants. Lastly, the system must meet the minimum ventilation requirements depending on the occupancy and square footage of the space. Provided the safety, pressurization, and thermal requirements in a space are satisfied, the VAV terminal or venturi valve would provide the minimum ventilation required to the space.

To meet the previously described requirements, the DCV system routinely samples air from each room through a duct probe located in an exhaust vent for each space. An additional duct probe monitors the supply air immediately following the VAV

terminal or venturi valve being supplied 100% outdoor air from the AHU. Through the duct probe, air is sent to a bank of sensors to determine if the VAV terminal or venturi valve needs make changes to system operation. The sensors used by the installed systems are given in Table 4. The TVOC and particulate sensors function to maintain the cleanliness or safety of the lab, as previously discussed. The airflow sensors maintain the required room pressurization and monitor the volume of air being supplied to the space. The temperature sensor helps the system maintain the thermal comfort in the space while the CO_2 sensor determines occupancy and allows the system to maintain the minimum ventilation required based on the real-time occupancy.

Table 4. Name and Description of System Sensors

Sensor Name	**Description**
CO_2	Concentration of CO_2 (ppm)
Temp	Temperature (°F)
TVOC	Total VOCs (ppm of isobutylene)
Small Particles	Number of airborne small particles (pcf)
CFM	Cubic Feet per Minute (cfm)

Data Analysis

The data derived from the sensors were analyzed to determine frequency, duration, and intensity of HVAC events. From this analysis, a typical week of HVAC events was defined. The frequency of events was determined by tabulating each occurrence throughout the research period. For each occurrence, the peak and average intensities were recorded. The average intensity was determined by summing the airflow intensity for each minute that airflow was greater than the baseline and then dividing by

the duration. A typical week was determined to be the average weekly frequency, intensity, and duration of HVAC events. Intensity is a weighted average of the average intensity of events for each week throughout the research period. This typical week derived from WSU laboratories is an acceptable estimate when applied to Air Force laboratories, which was the focus of this research, because the Air Force has a lower researcher density, similar HVAC set points, and similar laboratory functions.

Both Air Force and WSU researchers receive safety training in order to operate safely in the laboratory environment. However, Air Force laboratories do not typically have more than 20 researchers operating in the same laboratory space. An increased number of researchers in the same laboratory space increases the likelihood for an HVAC event. Specifically, more people generate greater amounts of CO_2 and increase the amounts of small particulates in the air from their activities. Therefore, relating to researcher density, Air Force laboratories should experience fewer HVAC events than WSU laboratories.

Air Force facilities are heated and cooled according to Unified Facility Criteria (UFC) 3-410-01, which asserts that comfort cooling is established at 78°F dry bulb and comfort heating is 68°F dry bulb (Department of Defense, 2013). According to Aircuity design documents, the HVAC system for the laboratories in each of the three WSU facilities are set to heat and cool to 74°F dry bulb. Therefore, when compared to Air Force setpoints, at the same relative humidity, the WSU system is working harder to cool because the threshold is 4°F lower and also working harder to heat because the target temperature is 6°F higher. The increased flexibility provided by the Air Force setpoints should produce fewer HVAC events for a similar system setup. When a DOAS unit is

used, the thermal requirements can be handled by the parallel system which will not affect DCV operation. As discussed previously, UFC 3-410-01 states that the DOAS can take on the space latent requirements if it is cost effective. This research assumes that the parallel HVAC system is designed to meet both sensible and latent requirements in the space while the DOAS specifically addresses the space ventilation requirements.

WSU laboratories serve many different departments and therefore have many different functions. Biological Science I Laboratories function as biomedical, clinical, biology, and physiology laboratory spaces. Diggs Laboratory contains neuroscience, genomics, biochemical, sedimentation, geochemistry and water chemistry laboratory spaces. Oelman Hall functions mostly as chemistry or earth and environmental science laboratories (WSU, 2014). These laboratory functions mimic many Air Force laboratory functions as explained in the following section. However, for each laboratory, a baseline is established based on the anticipated laboratory function. The baseline will be different for each laboratory function; yet, the typical response of the DCV system should be similar because the research process is the same. Therefore, the average data from WSU laboratories is an acceptable estimate for Air Force laboratories.

Phase II – Facility

In Phase II typical Air Force laboratory characteristics are discussed to establish a range of laboratory configurations considered in this analysis. The ASHRAE Standard 62.1 minimum ventilation rate equation is then used to determine the baseline supply air required for each laboratory configuration. The fan energy demand and overall HVAC energy demand is then calculated for the status quo and DCV conditions. The energy demand for the two alternatives is then used to determine demand reduction.

Air Force Laboratory Characteristics

The diverse research avenues and objectives of Air Force laboratories drive unique laboratory characteristics which makes it difficult to define a typical Air Force laboratory. However, analyzing the recently constructed Battlespace Environment Laboratory (BEL) at Kirtland AFB reveals some characteristics typical for a USAF laboratory. The BEL classifies eight different laboratory zones requiring unique HVAC consideration due to the type of work being performed. These eight zones are given in Table 5 along with the square footage of each zone and occupant data.

Table 5. BEL Laboratory HVAC Zones, Square Footage, and Occupant Data

Lab Name	Sq ft	# of Occupants	Occupants/1000 sq ft
Mass Spectrometer	1910	19.1	10
Electronics	890	8.9	10
LabCEDE	2200	20.4	9.27
Mumbo Jumbo	2812	28.1	9.99
BEC	1100	11	10
Choise	1240	12.4	10
Space Chemistry	2510	19.8	7.89
Plasma Chemistry	2360	18.8	7.97

The largest laboratory is Mumbo Jumbo at 2812 square feet (sq ft), the smallest laboratory is Electronics at 890 sq ft, and the average square footage for all of these laboratories is 1878 sq ft. This research analyzed rooms ranging from 800 to 3000 sq ft, which will account for all of these laboratories and most of the laboratories in the USAF inventory. Additionally, two zones have eight occupants per 1000 sq ft and the other six zones have ten occupants per 1000 sq ft. A conservative estimate for the USAF

laboratory occupancy rate is the higher ten occupants per 1000 sq ft which will be used for this analysis. Any partial occupant values will be rounded up to the nearest whole person.

The BEL HVAC system consists of two AHUs providing supply air at a constant volume to maintain eight ACH in each zone with the exception of the Electronics and BEC zones which are supplied at six ACH. Outdoor air is provided at 50% of the supply air rate (3 or 4 ACH). This system setup is similar to the setup at WSU where a single AHU is meeting the sensible and latent requirements in the space. The BEL laboratories also have fan coil units (FCU) to help meet the cooling requirements in the space.

As previously stated, UFC 3-410-01 was updated on 1 July 2013 and asserts that a DOAS must be used when the total outdoor air requirement exceeds 1000 cfm. Every laboratory configuration this research considers meets this requirement; therefore, the HVAC system design in this analysis is a DOAS unit in parallel with a multi-zone VAV system. This system design, shown earlier in Chapter II, enables the split of the sensible and latent requirements, when cost effective, and the ventilation requirement is met exclusively by the DOAS. Additionally, the system design has a fan exclusive to supply air and another exclusive to exhaust. Therefore, total fan energy consumption must account for the energy consumed by both fans. Energy recovery is not considered in this analysis as explained in Chapter II; therefore, there is no pressure loss due to an energy recovery wheel.

Minimum Ventilation

The ASHRAE Standard 62.1-2010, the ASHRAE ventilation standard referenced by UFC 3-410-01, minimum ventilation rate calculation applied to the standard

laboratories yields the minimum ventilation requirements. Table 6-1, *Minimum Ventilation Rates in Breathing Zone*, in the ASHRAE Standard 62.1-2010 provides values for the people and area outdoor air flow rates. The occupancy category that most closely relates to a USAF laboratory is "university/college laboratory" which yields R_p = 10 cfm/person and R_a = 0.18 cfm/square foot (ASHRAE, 2010c). These rates and the standard laboratory characteristics are used in Equation 9 to determine the minimum ventilation rate. Recall that this equation was presented in Chapter II as Equation 1. V_{bz} is the amount of outdoor air required in the breathing zone in cfm. The first term is the occupant related ventilation requirement where R_p is the required outdoor flow rate in cfm per person and P_z is the zone population. The second term in the equation is the building related ventilation requirement where R_a is the required outdoor flow rate in cfm per square foot (sq ft) and A_z is the square footage of the zone in sq ft.

$$V_{bz} = R_p P_z + R_a A_z \tag{9}$$

Once the minimum ventilation rate in the breathing zone was determined, the zone outdoor airflow (V_{oz}) can be calculated using Equation 10 by accounting for air distribution effectiveness (E_z) within the zone. Table 6-2, *Zone Air Distribution Effectiveness*, in ASHRAE standard 62.1-2010 provides a value for the air distribution effectiveness based on the configuration of the air distribution system. It is assumed that each zone has a ceiling supply for heating or cooling and a ceiling return. Because the DOAS provides thermally neutral air, this configuration yields a zone air distribution

effectiveness value equal to one (E_z = 1) which equates the zone outdoor airflow to the breathing zone outdoor airflow.

$$V_{oz} = \frac{V_{bz}}{E_z} \tag{10}$$

The minimum ventilation rate determined from the ASHRAE Standard 62.1 minimum ventilation rate equation was then compared to the minimum required supply air to maintain room pressurization at negative 100 cfm. This comparison is necessary because all of the supply air is being provided by the AHU. In the existing system setup, it is possible that the minimum ventilation rate will not fully satisfy the supply air requirement to maintain the desired room pressurization. If this shortage occurs, the greater supply air requirement determines the minimum supply air baseline. If the system was operating as a true DOAS, as considered by this research effort, then the parallel HVAC system would maintain the heating and pressurization requirements with recirculated air from elsewhere in the facility.

An additional consideration for the minimum ventilation rate is the limits of the equipment. As discussed earlier, Variable Frequency Drives (VFDs) can only reduce flow to 20% of the full flow at design conditions. Based on existing laboratory facilities with DCV systems and Aircuity system documents the maximum purge ACH rate varies from 12 to 16 ACH (Aircuity, 2012; Bell, 2008; Chan, Rahe, & Watch, 2012; Sharp,

2008). Based on these case studies and documents the maximum purge rate for this research is established at 15 ACH; therefore, 15 ACH is the full flow design condition.

HVAC Energy Demand

To determine DCV system fan energy demand, the average weekly HVAC event frequency, duration, and intensity were applied to the baseline supply air rate. The supply air fan demand when employing a DCV system was then compared to the status quo supply air demand to determine the demand reduction. Figure 10 shows an example of this comparison; the difference between the status quo and actual DCV operation lines is the amount of energy (in ACH) saved by the DCV system. The third line, calculated DCV, is how DCV operation is calculated in this effort. There is an inherent inaccuracy in this approach which over predicts fan energy savings by overlooking the inefficiencies associated with changing the flow up and down as required. However, the stepwise approach to determining DCV energy savings is also used by Nielsen and Drivsholm (2010).

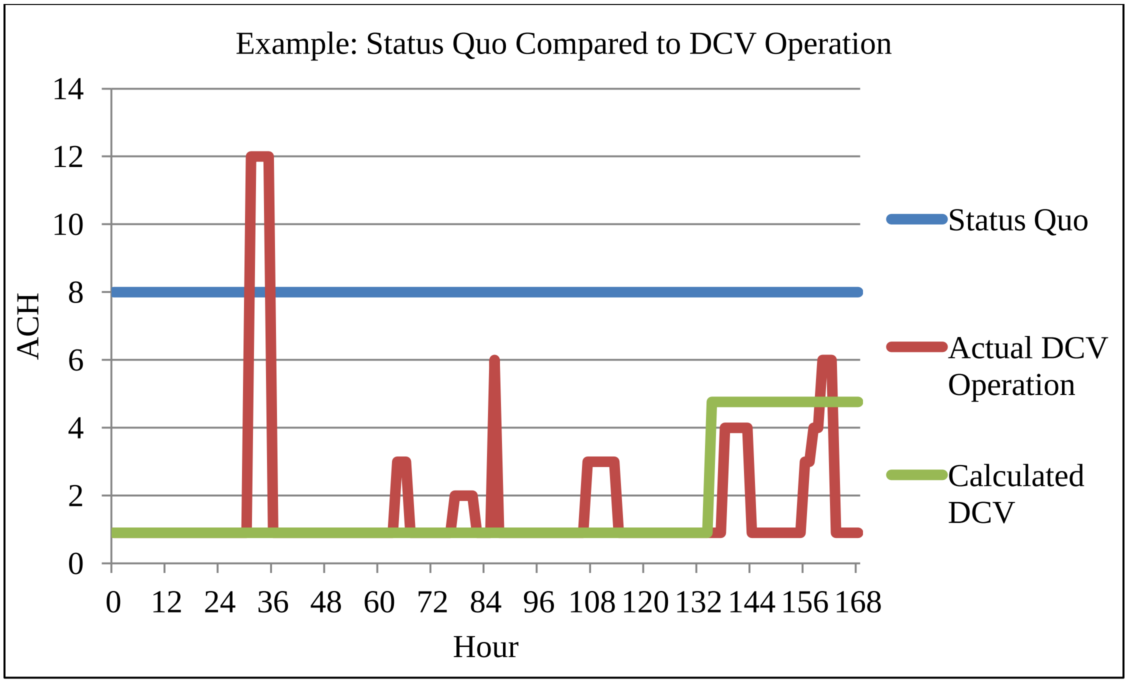

Figure 10. Example Weekly ACH Comparison

Using Equation 11, the weekly volumetric flow for each system is used to determine the Air Horsepower (AHP) delivered by the supply fan (Mleziva, 2010). *AHP* is the power consumed in horsepower (hp) for the given volumetric flow rate (Q) in cfm and total pressure (P) in in. w.g. Since the AHP is dependent on the density (ρ) of the air, standard air conditions, 68°F and 14.7 pounds per square inch (psi), are used throughout this analysis, which reduces the second term in Equation 11 to one (OSHA, 1999). Also required for the calculation is the total system pressure measured in inches water gauge (in. w.g.). Amon et al. (2007) asserted that most laboratories maintain total pressure set points greater than what is necessary to maintain acceptable conditions in the laboratory. In their tests, the existing supply total pressure set point is 3.1 in. w.g. and they determined that the optimal set point is 2 in. w.g. for their 137,025 sq ft laboratory

equipped with six AHUs at a total capacity of 228,000 cfm. Since the total pressure in a system can vary, the power demand was calculated using supply total pressure values from 1.0 to 6.0 in. w.g. in half inch increments. The WSU laboratory HVAC systems were designed based on a supply static pressure of 5 in. w.g. and an exhaust static pressure of 4.5 in. w.g. (Aircuity, 2012).

$$AHP = \frac{(Q)(P)}{6356} * \frac{\rho_{air}}{\rho_{standard\ air}} \tag{11}$$

The air horsepower is power consumed to push the air at the given flow rate and total pressure; yet, the total power consumed is greater due to inefficiencies in the motor, drives, and the fan itself, which have typical efficiencies of 90%, 94%, and 70%, respectively (Mleziva, 2010). Thus, the input power is the air horsepower divided by the efficiency of the system components, 59.2%; however, this only applies to the status quo system because varying the flow from the design condition in the DCV system will change the efficiencies as discussed earlier in Chapter II.

The equipment in the system limits the maximum reduction in flow to 20% of the full flow condition (Prachyl, 2010). Therefore, the minimum baseline is 20% of 15 ACH, or 3 ACH. At this reduced flow, using the fan cube law discussed in Chapter II, Equation 2, the load on the motor is reduced to 0.8% of the load at design conditions. Using Figure 11, originally published by Sfeir and Bernier (2005), the efficiency of a 15 – 25 hp motor at the baseline reduced flow (3 ACH) was determined to be 20%. This value is determined by reading the degradation factor from Figure 11 and then

multiplying by the motor full load efficiency, 90%.   A 15 – 25 hp motor is considered based on product documentation for a Munters DryCool™ Standard DOAS (Munters, 2011). Furthermore, the maximum flow from this unit is 16,000 cfm which can serve up to approximately 5000 sq ft, with 12 foot ceilings, while still being able to meet the maximum purge ACH rate. In a conversation with a Munters representative, a standard DOAS unit, using direct expansion, can only achieve a 50% reduction in flow; however, a non-standard unit using a chilled water system can achieve the desired reduction in flow (Munters, 2014). The worst case ACH increase when the DCV system responds is 5 ACH for the 4000 sq ft laboratory with five zones. In this state, using the cube fan law, the motor is operating at 3.7% of the full load. Using Figure 11, the efficiency of a 15 – 25 hp motor was determined to be 40%.

Similar to the motor, it is necessary to account for inefficiencies at part load for the Variable Frequency Drive (VFD). Using Figure 12 it is possible to estimate the efficiency of the VFD based on the new VFD ASHRAE (2000) plot (Sfeir & Bernier, 2005). As previously described in Equation 4, fan law states that speed is proportional to flow; thus, the percentage of nominal speed can be read as percentage of nominal flow. The baseline operates at a 20% reduction in flow and the worst-case DCV operation operates at a 33% reduction in flow. Therefore, the VFD at the baseline is estimated to be 91% efficient and during worst case DCV operation the VFD is estimated to be 94% efficient. Because the analysis considers the DCV system to operate as a stepwise function, the efficiencies are only needed for the baseline and average DCV response flow rates.

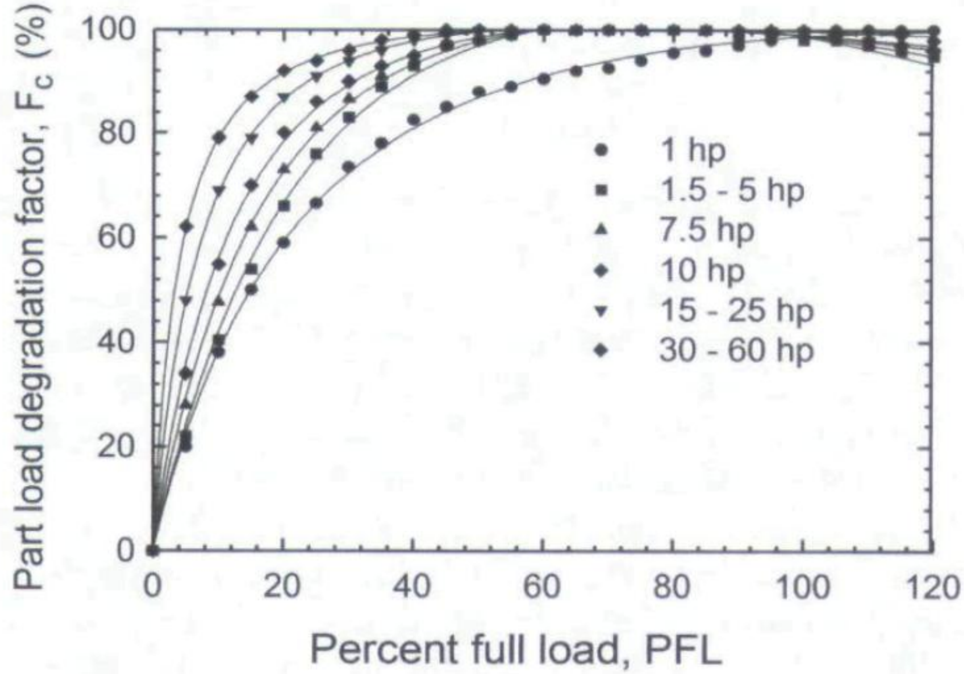

Figure 11. Part Load Motor Efficiencies (Sfeir & Bernier, 2005)

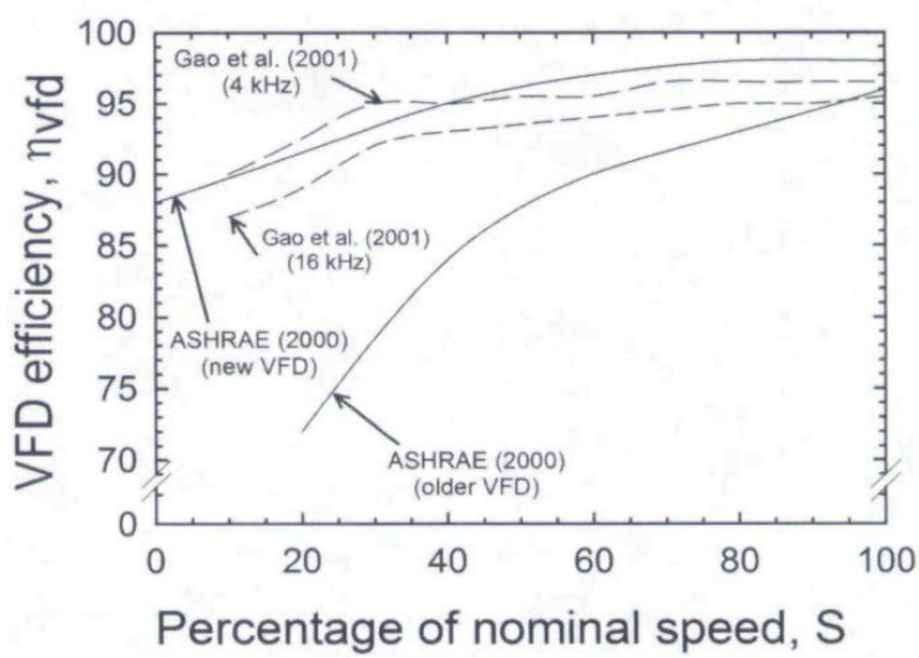

Figure 12. Part Load VFD Efficiency (Sfeir & Bernier, 2005)

Once calculated, the input power can then be converted to kilowatts (kW) and multiplied by the weekly operation time to determine energy consumption in kilowatt-hours (kWh). The weekly energy was then multiplied by 52 to determine the annual supply fan energy consumption. This same process is executed to determine exhaust fan energy demand; however, because this fan is pushing against the exhaust side of the HVAC system, the static pressure is 0.5 in. w.g. less than supply side (Aircuity, 2012). Additionally, the laboratories are maintained at 100 cfm negative pressure to ensure that any hazardous airborne substances are maintained within the room until exhausted outside.

Westphalen and Koszalinski (1999), in a report produced for the DOE, calculated the national parasitic energy consumption for commercial HVAC system. The analysis is based on the 1995 Commercial Building Energy Consumption Survey and used heating and cooling models developed by Lawrence Berkeley National Laboratory (LBNL). The models were based on engineering calculations and building site-measured data (Westphalen & Koszalinski, 1999). In the first volume of the report, completed two years later, Westphalen and Koszalinski follow the same methodology to calculate the national conditioning energy consumption.

In their report, Westphalen and Koszalinski (1999) assert that the total fan energy in an HVAC system accounts for approximately 85% of the total parasitic energy in the system. Parasitic energy is the energy used in the HVAC system to distribute conditioned air, discharge heat generated by cooling systems, and provide ventilation (Westphalen & Koszalinski, 1999). In a separate volume of the same report, Wesphalen and Koszalinski (2001), calculate that the parasitic energy consumption is approximately

one-third of the total HVAC energy consumption. Furthermore, the results of their study were compared to five other similar studies with equivalent results. The parasitic energy portion for four of the other five studies constituted a lower percentage of the total energy consumed. The minimum and maximum values reported by each study for each category maintain a parasitic energy percentage less than one-third of the total overall HVAC energy consumed (Westphalen & Koszalinski, 2001). Additional literature supports that parasitic energy is approximately one-third of total HVAC energy consumption (e.g. Brendel, 2010; Knight, 2012; Perez-Lombard, Ortiz, & Maestre, 2011). Specific to this research, the parasitic energy at the three WSU laboratory facilities accounted for 19% to 25% of the total HVAC energy consumed when the fan energy is 85% of the total parasitic energy (Aircuity, 2012). Therefore, after the supply and exhaust fan energy is calculated, the total HVAC energy consumption is determined where fan energy is 85% of the parasitic energy and parasitic energy is one-third of the total HVAC energy.

Phase III - Economic Analysis

The first step in phase III was to analyze USAF laboratories of varying size using the results of phases I and II to determine total HVAC energy savings using DCV. The energy demand was then priced at the United States average commercial electricity rate to determine the amount of cost savings achieved utilizing a CO_2-based DCV system. An economic analysis was then performed to determine the life-cycle cost effectiveness of the DCV system.

BLCC Inputs

The economic analysis for this research was performed using the Building Life-Cycle Cost 5 (BLCC) program developed by the National Institute of Standards and

Technology (NIST). BLCC was chosen because Military Construction (MILCON) energy projects are required to use Department of Energy (DOE) energy escalation rates and indexes (USAF, 2011). The BLCC program is a tool provided by the DOE for the analysis of energy projects and incorporates the required escalation rates and indexes. The life-cycle cost method was chosen because it compares the two alternatives over the entire life of the system. This approach considers the economic advantages and disadvantages of each system when performing the analysis.

Each alternative requires energy, capital, and operations and maintenance related inputs to perform the analysis. Energy inputs are broken into annual consumption in kilowatt hours (kWh), price per kWh, annual demand charge, and annual utility rebate. This analysis used the DOE Energy Efficiency and Renewable Energy (EERE) office published average federal electricity price of $0.06 per kWh (DOE: EERE, 2013). This price includes the cost of demand charges; thus, the annual demand charge input was not used. Further, annual utility rebates vary significantly depending on location and were not considered. Any utility rebate should be considered additional savings.

Capital inputs for the analysis include the initial cost of the alternative, expected life, and residual value factor. The initial cost is the initial cost of the DCV system. This was calculated by averaging the per square foot cost for each of the three WSU facilities. The initial cost of the HVAC system was not considered since it is the same for both conditions. The expected life of the system is how long the system will function before requiring replacement. ASHRAE created an online service life survey in which individuals may submit data regarding their HVAC system life and maintenance costs. The average life for an AHU still in service providing VAV at variable temperature

averages 20.1 years (ASHRAE, 2014a). Therefore, 20 years will be used as a conservative estimate for expected life in this research effort. The last capital input is the residual value factor or the system's worth at the end of the expected life. This analysis assumed that the residual value is zero because the government does not generally receive any value from an HVAC system that is beyond its useful life.

The last input required for analysis is the operations and maintenance cost for each alternative. Using the same ASHRAE survey data, with 267 buildings reporting, the average HVAC maintenance cost is $0.336 per square foot. Only three laboratory facilities reported data with an average maintenance cost of $0.667 per square foot (ASHRAE, 2014b). It is expected that laboratory have higher HVAC maintenance costs due to the higher demand on laboratory HVAC equipment. However, a conservative estimate is that the annual HVAC maintenance costs are the same for both the status quo and DCV systems. The DCV system itself requires additional annual maintenance to maintain the accuracy of the sensors and ensure proper system operation. The annual maintenance costs for the DCV alternative will be calculated based on the projected costs for the WSU systems. For each of the WSU DCV systems, the maintenance costs are the same.

BLCC Calculations

The BLCC5 software calculations are all based the present value (PV) of costs determined by discounting. The analysis uses mid-year discounting which assumes that the entire cash flow for a given year occurs at the midpoint of that year. The real discount rate for 2013 is 3% as published by NIST in the Annual Supplement to NIST Handbook 135 (NIST, 2013). All future costs are discounted at 3% to determine the PV

of the cost. Once all costs are in PV, the savings and expenses for an alternative are summed to determine the life-cycle cost. The analysis excludes inflation and all costs are given in constant dollars. In addition to the life-cycle cost of an alternative, the following economic measures are also used: net savings, savings-to-investment ratio (SIR), adjusted internal rate of return (AIRR), and payback.

The net savings is calculated when comparing two alternatives to determine the total savings achieved by one alternative over another. The net savings of an alternative is calculated by subtracting the life-cycle cost of the alternative from the life-cycle cost of the status quo. The numerator in the SIR is the PV of the status quo costs minus the PV of the alternative costs, which is the amount saved by the DCV system. The denominator in the SIR is the PV of the additional investment required for the DCV system (Fuller, Rushing, & Meyer, 2001). When this value is one or greater, the system achieves a net savings. In this analysis, the SIR was used to determine the maximum initial cost of the DCV system. This was achieved by performing a comparative analysis between the status quo and DCV system when the DCV system had no initial cost. The savings achieved by the DCV system under these conditions is equal to the maximum initial cost of the DCV system to achieve net savings.

The AIRR is used to make decisions and prioritize projects by calculating investment performance. Equation 12 shows how the AIRR is calculated for an alternative. The variable r is the real discount rate (3%) and N is the lifespan of the alternative (20 years) (Fuller, Rushing, & Meyer, 2001).

$$AIRR = (1 + r) * SIR^{\left(\frac{1}{N}\right)} - 1 \qquad (12)$$

The payback for an investment can be calculated with or without discounting. Simple payback is calculated without discounting future cash flows while discounted payback does discount future cash flows. The payback calculates the number of years required for savings, discounted or not, to at least equal the additional investment costs. The quicker payback is achieved, the stronger the investment; however, payback calculation do not consider cash flows after payback has been achieved. Therefore, payback should be used in conjunction with another economic measure before a decision is made.

BLCC Outputs

The BLCC5 program performs its analysis and outputs a comparative analysis report showing if the alternative is cost effective. The comparison is performed in PV costs. The report shows base case, alternative, and savings value for each cost category. Figure 13 is an example PV comparison report output. The report then outlines the net savings value for the alternative when compared to the status quo, the SIR, the AIRR, and the payback period. The energy savings and emissions reduction for each case is also detailed. Figure 14 is an example of the savings and emissions results presented in the remainder of the report.

Comparison of Present-Value Costs
PV Life-Cycle Cost

	Base Case	Alternative	Savings from Alternative
Initial Investment Costs:			
Capital Requirements as of Base Date	$0	$160,000	-$160,000
Future Costs:			
Energy Consumption Costs	$392,138	$0	$392,138
Energy Demand Charges	$0	$0	$0
Energy Utility Rebates	$0	$0	$0
Water Costs	$0	$0	$0
Routine Recurring and Non-Recurring OM&R Costs	$16,381	$107,600	-$91,219
Major Repair and Replacements	$0	$0	$0
Residual Value at End of Study Period	$0	$0	$0
	------------	------------	------------
Subtotal (for Future Cost Items)	$408,519	$107,600	$300,919
	------------	------------	------------
Total PV Life-Cycle Cost	$408,519	$267,600	$140,919

Figure 13. BLCC5 Comparative Analysis Report - Comparison of PV Costs

Net Savings from Alternative Compared with Base Case

PV of Non-Investment Savings	$300,919
- Increased Total Investment	$160,000

Net Savings	$140,919

Savings-to-Investment Ratio (SIR)

SIR = 1.88

Adjusted Internal Rate of Return

AIRR = 6.31%

Payback Period

Estimated Years to Payback (from beginning of Beneficial Occupancy Period)

Simple Payback occurs in year 8

Discounted Payback occurs in year 9

Energy Savings Summary

Energy Savings Summary (in stated units)

Energy	-----Average	Annual	Consumption-----	Life-Cycle
Type	Base Case	Alternative	Savings	Savings
Electricity	423,188.0 kWh	0.0 kWh	423,188.0 kWh	8,462,601.4 kWh

Energy Savings Summary (in MBtu)

Energy	-----Average	Annual	Consumption-----	Life-Cycle
Type	Base Case	Alternative	Savings	Savings
Electricity	1,444.0 MBtu	0.0 MBtu	1,444.0 MBtu	28,875.6 MBtu

Emissions Reduction Summary

Energy	-----Average	Annual	Emissions-----	Life-Cycle
Type	Base Case	Alternative	Reduction	Reduction
Electricity				
CO2	365,605.26 kg	0.00 kg	365,605.26 kg	7,311,104.22 kg
SO2	2,919.38 kg	0.00 kg	2,919.38 kg	58,379.63 kg
NOx	681.09 kg	0.00 kg	681.09 kg	13,619.88 kg
Total:				
CO2	365,605.26 kg	0.00 kg	365,605.26 kg	7,311,104.22 kg
SO2	2,919.38 kg	0.00 kg	2,919.38 kg	58,379.63 kg
NOx	681.09 kg	0.00 kg	681.09 kg	13,619.88 kg

Figure 14. BLCC5 Comparative Analysis Report - Savings and Emissions Reduction

Additional outputs for the BLCC5 program include a cash flow analysis and summary LCC report. The cash flow analysis report details all of the costs associated with each alternative throughout the service life. The report presents the capital investment, operating cost, and total cash flow for each alternative. Figure 15 shows an

example total cash flow for the DCV alternative. Shown in the cash flow is the initial cost annual maintenance of the DCV system. All other HVAC costs are the same for each alternative; therefore, they are not included. The cash flows are integral because they establish the foundation for the LCC and comparative analysis. The summary LCC report, Figure 16, shows the present and annual value for each of the costs associated with each alternative. Also included is the total LCC for each alternative. Many categories are zero because the associated costs have been included elsewhere or are the same for each alternative. The status quo alternative has non-annually recurring costs equal to the first three years of DCV maintenance costs because the maintenance costs for the first three years is included in the initial price. These additional outputs provide the data necessary to determine the comparative analysis results and provide a better understanding of the costs for each option.

Sum of All Cash Flows

Year Beginning	Capital	OM&R	Total
Jan 2014	$160,000	$5,460	$165,460
Jan 2015	$0	$5,624	$5,624
Jan 2016	$0	$5,793	$5,793
Jan 2017	$0	$5,966	$5,966
Jan 2018	$0	$6,145	$6,145
Jan 2019	$0	$6,330	$6,330
Jan 2020	$0	$6,520	$6,520
Jan 2021	$0	$6,715	$6,715
Jan 2022	$0	$6,917	$6,917
Jan 2023	$0	$7,124	$7,124
Jan 2024	$0	$7,338	$7,338
Jan 2025	$0	$7,558	$7,558
Jan 2026	$0	$7,785	$7,785
Jan 2027	$0	$8,018	$8,018
Jan 2028	$0	$8,259	$8,259
Jan 2029	$0	$8,507	$8,507
Jan 2030	$0	$8,762	$8,762
Jan 2031	$0	$9,024	$9,024
Jan 2032	$0	$9,295	$9,295
Jan 2033	$0	$9,574	$9,574
------------	------------	------------	------------
Total	$160,000	$146,713	$306,713

Figure 15. Example Total Cash Flow Report

Alternative: Status Quo
LCC Summary

	Present Value	Annual Value
Initial Cost Paid By Agency	$0	$0
Energy Consumption Costs	$392,138	$26,360
Energy Demand Costs	$0	$0
Energy Utility Rebates	$0	$0
Water Usage Costs	$0	$0
Water Disposal Costs	$0	$0
Routine Annually Recurring OM&R Costs	$0	$0
Routine Non-Annually Recurring OM&R Costs	$16,381	$1,101
Major Repair and Replacement Costs	$0	$0
Less Remaining Value	$0	$0
	------------	------------
Total Life-Cycle Cost	$408,519	$27,462

Alternative: DCV
LCC Summary

	Present Value	Annual Value
Initial Cost Paid By Agency	$160,000	$10,756
Energy Consumption Costs	$0	$0
Energy Demand Costs	$0	$0
Energy Utility Rebates	$0	$0
Water Usage Costs	$0	$0
Water Disposal Costs	$0	$0
Routine Annually Recurring OM&R Costs	$107,600	$7,233
Routine Non-Annually Recurring OM&R Costs	$0	$0
Major Repair and Replacement Costs	$0	$0
Less Remaining Value	$0	$0
	------------	------------
Total Life-Cycle Cost	$267,600	$17,989

Figure 16. Example Summary LCC Report

Summary

This research effort followed a methodology which analyzed DCV system data at WSU to determine the duration, frequency, and intensity of HVAC events. A typical

week of HVAC events was determined and combined with the minimum supply air baseline calculated in phase II to yield weekly supply air demand. This demand was converted to HVAC energy cost and compared to the HVAC energy cost of the status quo. The difference between the two energy costs is the savings achieved by the DCV system. An economic analysis using life-cycle costing, saving-to-investment ratio, and discounted payback techniques was then performed using the BLCC5 software to determine the economic feasibility of the CO_2-based DCV system.

IV. Analysis and Results

In this chapter, the results from each phase of the methodology are presented. Phase I analyzed the demand controlled ventilation (DCV) systems installed on three Wright State University (WSU) laboratory facilities to determine a typical week of heating, ventilation, and air-conditioning (HVAC) events. Additionally, the data shows that the DCV system is able to maintain the indoor environment within safe limits for the monitored parameters. Phase II discusses typical United States Air Force laboratory characteristics and explains how those characteristics are used in the analysis. The minimum ventilation requirements are calculated for the range of laboratory configurations. Based on the minimum ventilation requirements, the fan energy and overall HVAC energy is calculated. Lastly, phase III synthesizes the results from phases I and II to determine the energy savings achieved using the DCV system. The energy consumption results are then used to complete a life-cycle cost analysis (LCCA).

Phase I

During Phase I, the data from three WSU laboratory facilities were analyzed to define a typical week of HVAC events as determined by the frequency, duration, and intensity of the events. An HVAC event exists when the system supplies greater than 50 cubic feet per minute (cfm) of ventilation above the baseline. The results of this phase of the analysis are supported by Sharp (2010), who asserted that approximately 99% of the time a laboratory can maintain IAQ at a reduced air change (ACH) rate.

Typical Week

The following table shows the frequency of HVAC events by room by week and a four week average by room. Table 6 shows the frequency of HVAC events for the WSU

laboratories. The facility average for Biological Science I is 3.9 events per week yet, 61% of the rooms have a 4-week average below the facility average. Although Diggs Laboratory has half as many rooms as Biological Science I Laboratory it averaged 15.75 events per week. Furthermore, 44% of the rooms exceeded the 4-week facility average, this is a more balanced dispersion of values when compared to Biological Science I. Oelman Hall has 11 rooms and a 4-week facility average of 5.39 events per week. Similar to Biological Science I Laboratory, there is one room that has a significantly higher frequency of HVAC events that the other rooms. However, most rooms have a frequency near the facility average and 45% of the rooms are above the facility average.

The standard deviation for HVAC event frequency is 4.74, 19.81, and 7.31 for Biological Science I, Diggs, and Oelman Hall, respectively. For each facility, the standard deviation is greater than the four-week average and the standard deviation range about the average includes zero. For each facility there are several rooms with a greater frequency which increases the standard deviation. Specifically, rooms 123 and 17 in Biological Science I, rooms 25, 104, 165, and 204 in Diggs Laboratory, and room 443 in Oelman Hall.

Averaging the 4-week averages for the 38 rooms in all three laboratories yields an overall frequency average of 7.1 HVAC events per week with a standard deviation of 11.85. Even though the standard deviation is greater than the average, when compared to the overall frequency average, 30 of the 38 rooms (79%) have a 4-week frequency average below the overall average. Specifically, 83% of the rooms in Bio I, 56% of the rooms in Diggs, and 91% of the rooms in Oelman Hall are below the overall average.

Table 6. WSU HVAC Event Frequency

Biological Science I						Diggs Laboratory						Oelman Hall					
Room	Week 1	Week 2	Week 3	Week 4	4-Week Avg	Room	Week 1	Week 2	Week 3	Week 4	4-Week Avg	Room	Week 1	Week 2	Week 3	Week 4	4-Week Avg
11	2	5	3	2	3	25	65	9	6	13	23.25	431	3	3	7	15	7
14	3	3	4	0	2.5	104	18	16	17	16	16.75	433	3	3	7	15	7
17	16	5	7	10	9.5	125	7	8	4	7	6.5	443	11	4	23	32	17.5
18	4	11	9	6	7.5	135	5	1	0	0	1.5	450	2	0	1	5	2
19	8	8	4	1	5.25	165	30	22	24	44	30	453	2	0	4	16	5.5
22	6	4	2	0	3	204	59	56	64	45	56	455	0	1	1	12	3.5
28	2	2	1	2	1.75	225	4	0	1	0	1.25	456	0	0	3	17	5
119	2	5	5	6	4.5	235	5	1	0	0	1.5	458	0	0	2	21	5.75
123	23	21	13	14	17.75	265	3	10	6	1	5	463	0	2	0	7	2.25
128	0	0	0	0	0							469	0	1	3	3	1.75
203	3	3	1	0	1.75							471	0	0	3	5	2
209	7	2	2	6	4.25												
210	4	0	3	0	1.75												
215	8	2	8	4	5.5												
216	0	0	2	3	1.25												
220	0	1	0	1	0.5												
226	0	0	0	0	0												
230	1	1	0	1	0.75												

The duration of HVAC events also vary by room and by facility. The following tables show the average duration of HVAC events in minutes by room by week as well as the 4-week average. Table 7 shows the duration of WSU laboratory HVAC events. Biological Science I room 216 experienced one constant HVAC event for the entirety of the four week research period. This is an anomaly because the room does not reach equilibrium at the baseline airflow. Biological Science I room 216 should have a baseline at 550 cfm greater than what is established because at this new baseline the room is at equilibrium 98.6% of the time. Re-analyzing this room with the updated baseline yields a 4-week average frequency of 1.25 and a 4-week average duration of 115.4 minutes. This updated data was used throughout the remainder of this research. The facility duration average for Biological Science I, using the updated data for room 216, is 25.26 minutes. The facility duration average for Diggs Laboratory is 16.24 minutes while the facility duration average for Oelman Hall is 9.08 minutes. Oelman Hall did not experience prolonged HVAC events in most of the rooms and has only one room with an average greater than 9.8 minutes which is room 443 at 27.02 minutes.

Using the revised values for Biological Science I room 216, the new weekly average frequency remains unchanged at 7.1 events per week and the updated average duration is 18.44 minutes. The total average time for an HVAC event per week is 130.92 minutes or 1.3% of the time in a week. The annual time above the baseline is 4 days 17 hours and 28 minutes.

Table 7. WSU HVAC Event Duration

Biological Science I Laboratory							Diggs Laboratory							Oelman Hall					
Room	Week 1	Week 2	Week 3	Week 4	4-Week Avg		Room	Week 1	Week 2	Week 3	Week 4	4-Week Avg		Room	Week 1	Week 2	Week 3	Week 4	4-Week Avg
11	6.00	12.80	21.33	55.50	23.91		25	29.18	11.00	11.00	17.54	17.18		431	11.67	11.33	7.14	9.00	9.79
14	12.00	25.00	37.75	0.00	18.69		104	10.11	14.94	11.35	11.81	12.05		433	11.67	11.33	7.14	9.00	9.79
17	33.19	40.60	47.71	39.00	40.13		125	116.00	9.75	8.75	7.14	35.41		443	17.91	27.25	30.83	32.09	27.02
18	25.50	33.73	35.89	21.13	29.06		135	31.00	14.00	0.00	0.00	11.25		450	5.50	0.00	7.00	3.40	3.98
19	32.00	18.63	20.25	13.00	20.97		165	8.27	8.14	12.67	14.59	10.92		453	8.00	0.00	11.00	16.81	8.95
22	21.33	13.25	13.00	0.00	11.90		204	28.63	28.18	21.09	38.71	29.15		455	0.00	12.00	5.00	7.00	6.00
28	19.50	19.00	13.00	32.00	20.88		225	12.75	0.00	13.00	0.00	6.44		456	0.00	0.00	12.33	13.41	6.44
119	6.50	25.20	23.20	17.17	18.02		235	31.00	14.00	0.00	0.00	11.25		458	0.00	0.00	13.50	19.10	8.15
123	25.04	73.24	28.31	31.07	39.42		265	8.67	12.30	9.17	20.00	12.53		463	0.00	17.00	0.00	7.71	6.18
128	0.00	0.00	0.00	0.00	0.00									469	0.00	11.00	9.80	9.33	7.53
203	12.67	12.33	5.00	0.00	7.50									471	0.00	0.00	10.67	13.40	6.02
209	29.57	51.50	76.50	21.00	44.64														
210	41.25	0.00	43.00	0.00	21.06														
215	23.38	25.00	23.75	19.25	22.84														
216	0.00	0.00	98.00	132.80	115.40														
220	0.00	39.00	0.00	13.00	13.00														
226	0.00	0.00	0.00	0.00	0.00														
230	5.00	12.00	0.00	12.00	7.25														

The intensity of the HVAC events is the last piece of information required to define an average week for HVAC events. Table 8 shows the weighted average intensity in cfm by room by facility for the WSU laboratories. The facility average for Biological Science I is 406 cfm over the baseline per event while the facility average for Diggs Laboratory is 321 CFM over the baseline per event. This is lower than Biological Science I Laboratory because even though Diggs Laboratory experienced a greater number of events each of the events was not as intense. Further, three of the nine rooms in Diggs Laboratory experienced an average intensity less than 50 cfm greater than the baseline. The facility average for Oelman Hall is 164 cfm over the baseline per event. Oelman Hall experienced a reduced frequency and intensity of HVAC events when compared to the other two facilities. Only one room in Oelman Hall experienced an average intensity greater than 170 cfm over the baseline with room 443 at 784 cfm over the baseline per event.

The overall average HVAC event intensity is 316 cfm above the baseline per event. Therefore, an average week for a single zone has 7.1 HVAC events each lasting 18.44 minutes at an intensity of 316 cfm above the baseline per event. However, this typical week data is contingent upon the system being able to maintain a safe indoor air quality (IAQ) for the laboratories.

Table 8. WSU HVAC Event Intensity

Biological Science I Laboratory						Diggs Laboratory						Oelman Hall					
Room	Week 1	Week 2	Week 3	Week 4	4-Week Average	Room	Week 1	Week 2	Week 3	Week 4	4-Week Average	Room	Week 1	Week 2	Week 3	Week 4	4-Week Average
11	59	109	343	667	294	25	459	432	552	517	490	431	78	154	233	202	167
14	108	374	546	0	257	104	835	707	419	352	578	433	78	154	233	209	169
17	205	1277	1473	271	806	125	142	151	200	151	161	443	529	579	1114	914	784
18	1313	1400	625	434	943	135	83	95	0	0	44	450	53	0	52	123	57
19	455	821	386	191	463	165	236	236	333	526	333	453	61	0	114	171	87
22	157	638	163	0	239	204	898	823	785	921	857	455	0	55	60	97	53
28	204	570	245	138	289	225	79	0	100	0	45	456	0	0	230	157	97
119	196	162	465	253	269	235	83	95	0	0	44	458	0	0	491	186	169
123	540	574	915	1136	791	265	289	364	151	524	332	463	0	318	0	133	113
128	0	0	0	0	0							469	0	59	99	65	56
203	391	132	125	0	162							471	0	0	101	119	55
209	603	312	489	348	438												
210	739	0	143	0	221												
215	165	220	1466	533	596												
216	540	540	569	584	558												
220	0	247	0	123	93												
226	0	0	0	0	0												
230	244	1577	0	1740	890												

Laboratory Safety

As previously discussed, the DCV system monitors CO_2, small particulates, and total volatile organic compound (TVOC) data for each of the laboratory spaces to maintain a safe IAQ. The American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) Standard 62.1-2010 appendix C asserts that CO_2 concentrations not greater than 700 ppm above the outdoor CO_2 concentrations will satisfy the majority of visitors to the space with respect to human bioeffulents. Additionally, acceptable outdoor air concentrations range from 300 ppm to 500 ppm (ASHRAE, 2010c). Therefore, if the indoor CO_2 concentrations are maintained below 1000 ppm the IAQ with respect to human bioeffulents will satisfy the majority of visitors. Figure 17 shows the maximum and average CO_2 concentrations by room by week for the Biological Science I Laboratory. The maximum value the Bio Science I Laboratories experience occurs in room 17 during the first week and is 837 ppm, which is within the accepted levels published by ASHRAE. Additionally, it can be seen that each room maintains an average CO_2 concentration slightly less than 400 ppm, which is the outdoor CO_2 concentration shown by the air handler unit (AHU) reading.

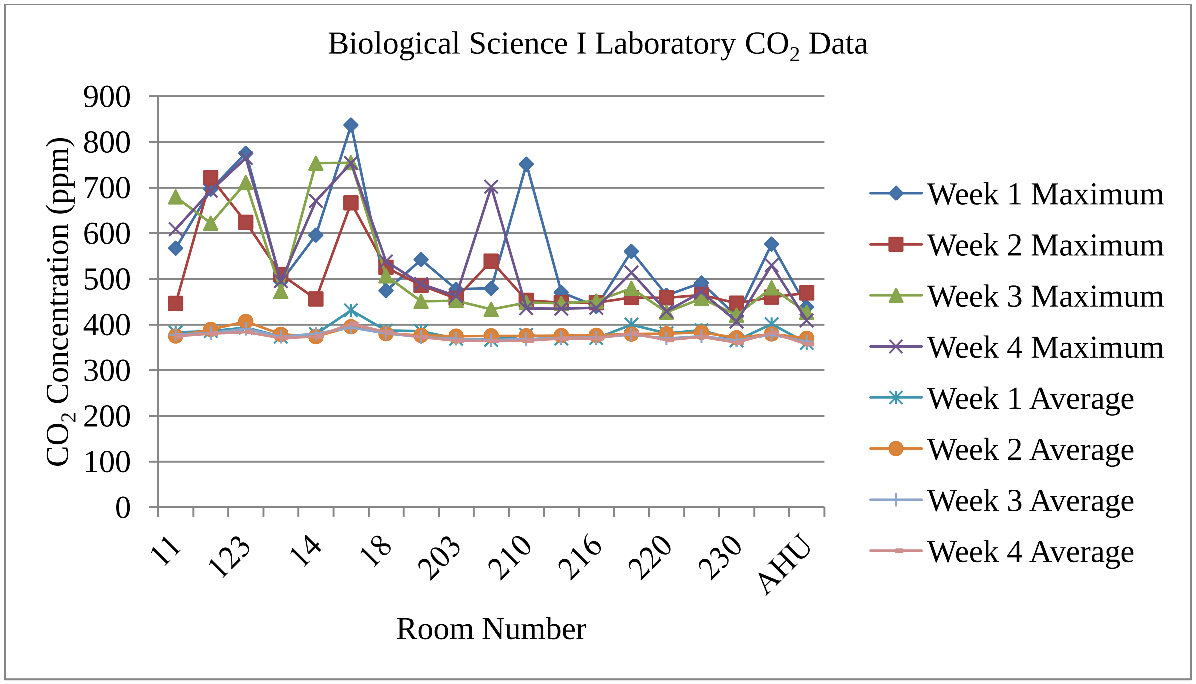

Figure 17. Biological Science I Laboratory Maximum and Average CO_2 Concentrations

Figure 18 shows the same CO_2 concentration data for Diggs Laboratory. The maximum value achieved is less than what is experienced in Biological Science I and is therefore within accepted limits. The average CO_2 concentration is maintained slightly above 400 ppm but less than or equal to the AHU average, which means that the indoor environment is less concentrated with CO_2 than the outdoor environment.

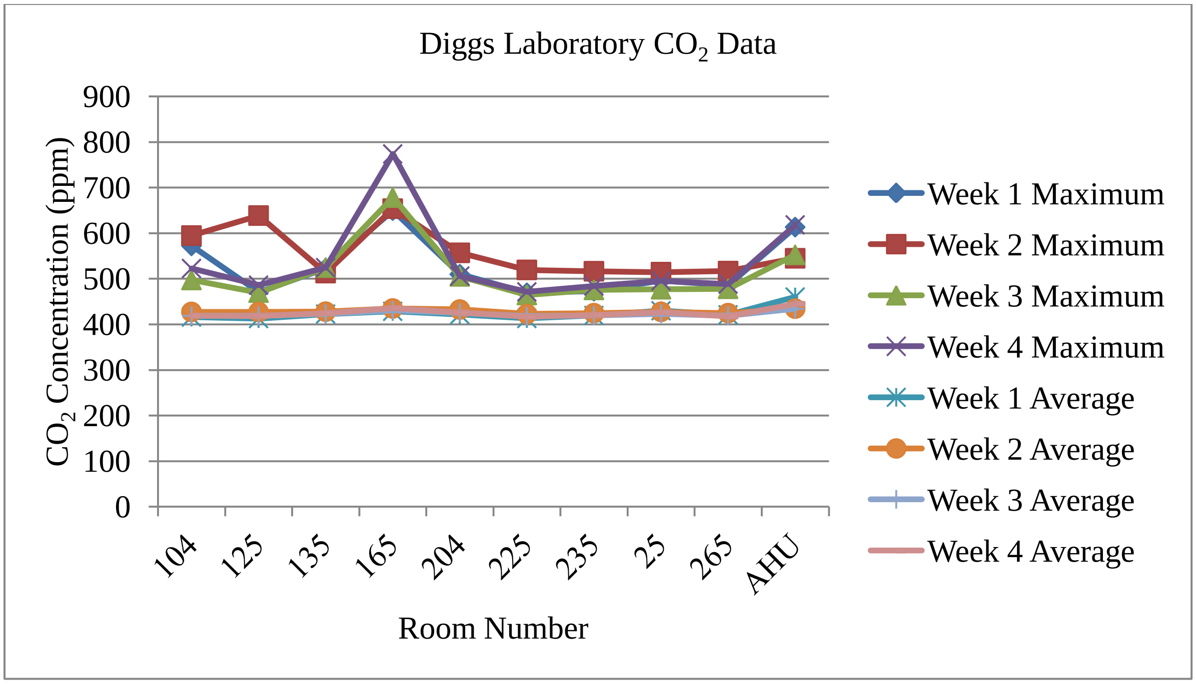

Figure 18. Diggs Laboratory Maximum and Average CO_2 Concentrations

Figure 19 shows the CO_2 concentration data for Oelman Hall. Again, the DCV system is able to maintain CO_2 concentrations within the approved ASHRAE limits. The average CO_2 concentration for Oelman Hall is maintained slightly above the outdoor concentration given by the AHU average; yet, the approximately 50 ppm difference would not be noticed.

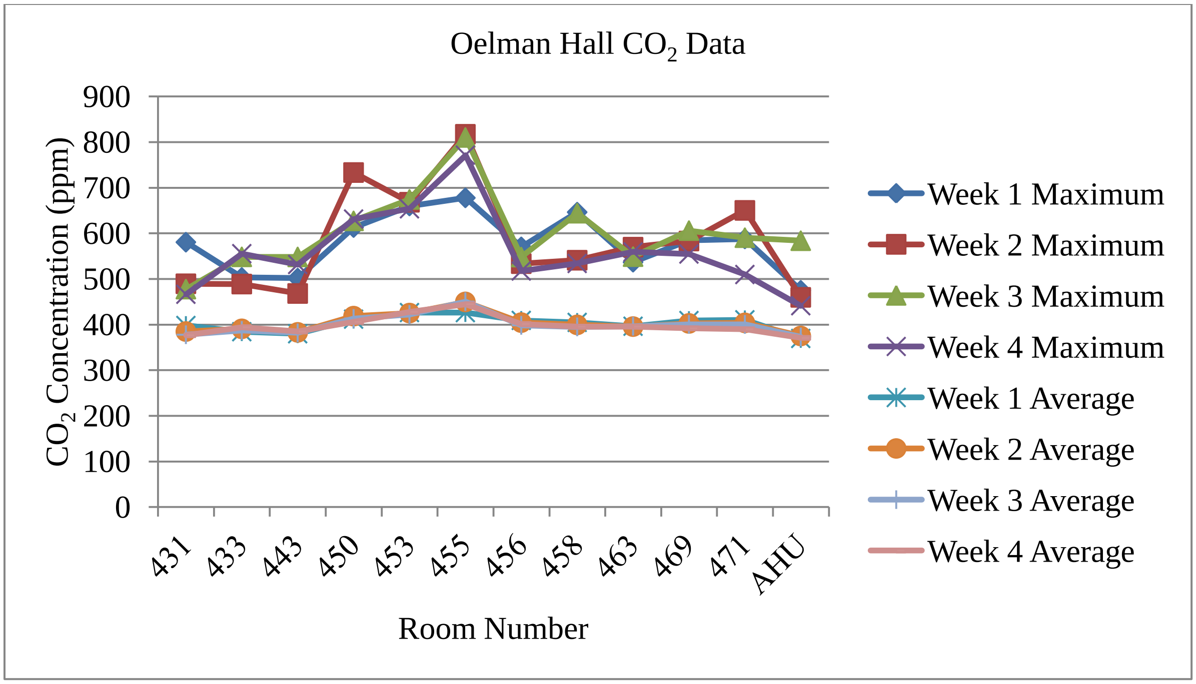

Figure 19. Oelman Hall Maximum and Average CO_2 Concentrations

The average AHU data from each of the facilities suggests that the outdoor CO_2 concentration is approximately 400 ppm which increases the maximum accepted indoor concentration to 1100 ppm. The DCV system is able to maintain acceptable CO_2 levels in each laboratory space throughout the research period. These results suggest that visitors to the space will find the IAQ acceptable with regards to human bioeffluent production.

Currently there is not an indoor standard for small particulate limits; however, the Environmental Protection Agency's National Ambient Air Quality Standard for particulate matter, revised in 2013, establishes an ambient air limit for particulate matter less than 2.5 micrometers in diameter of 12 $\mu g/m^3$ and a 24-hour exposure limit of 35 $\mu g/m^3$ (Esworthy, 2013). Specifically measured by the system sensors are fine particles

ranging from 0.3 – 2.5µm in diameter. The data is reported in particles per cubic foot (pcf) which requires a conversion of units for comparison; however, the particulate composition and weight is unknown which prevents the conversion for comparison. Rosenthal and Brown (2014) assert that a typical indoor environment has about 1.5 million particles greater than 0.3µm per cubic foot. However, there are many factors that can affect ambient particle count.

The following figures show the average small particle count to allow a comparison between the AHU, or outdoor, reading and the room readings. Figure 20 shows the Biological Science I Laboratory small particle data. For each week, the average small particulate count at the AHU is greater than in the rooms.

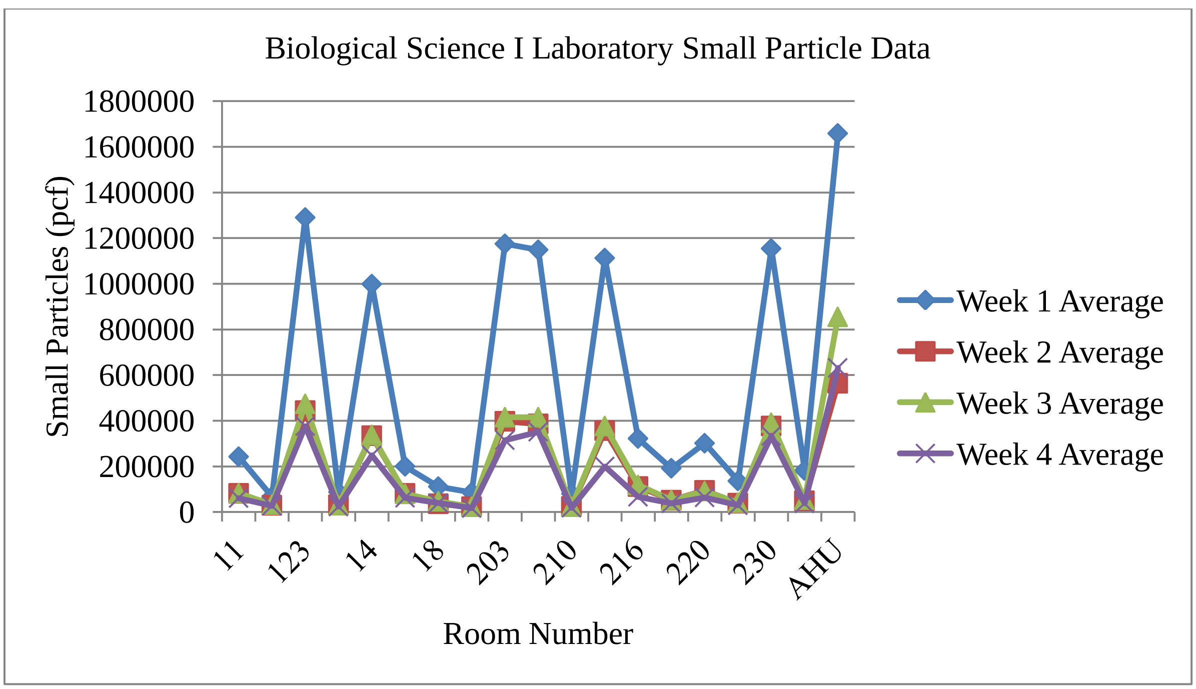

Figure 20. Biological Science I Laboratory Average Small Particle Concentrations

Diggs Laboratory small particle data, shown in Figure 21, is significantly lower than the Biological Science I Laboratory data; yet, the data follows the same trend. Week 1 data is the highest for both facilities while weeks two through four track very closely together. The AHU readings for Diggs Laboratory are approximately the same as the room readings, indicating that indoor concentrations are approximately the same as outdoor small particle concentrations.

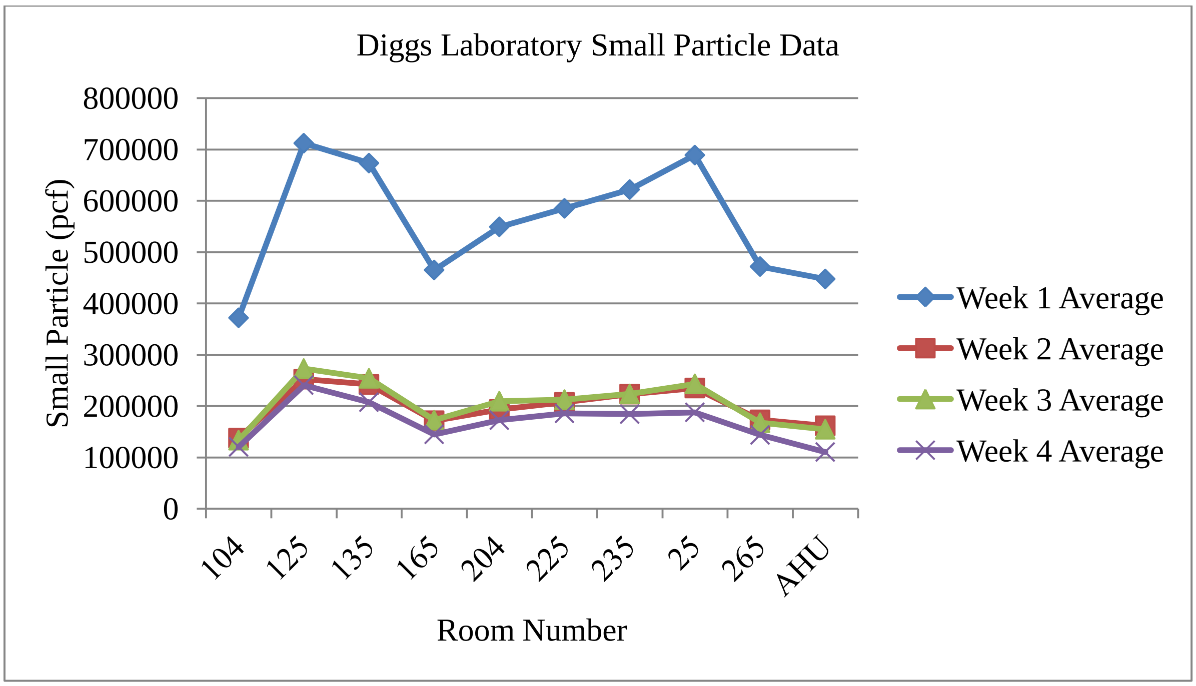

Figure 21. Diggs Laboratory Average Small Particle Concentrations

Figure 22 shows the small particle data for Oelman Hall during the research period. The data follows the same trends as the previously described facilities and tracks closely to Biological Science I Laboratory in magnitude. Week 1 again shows greater concentrations than the other weeks. The AHU readings are greater than the room

readings each week with the exception of room 443 during week three and rooms 433 and 443 during week four. For each facility, the average small particle room concentration does not exceed 1.4 million pcf, which is below what is often considered typical. Therefore, the system can maintain a safe lab environment regarding small particles.

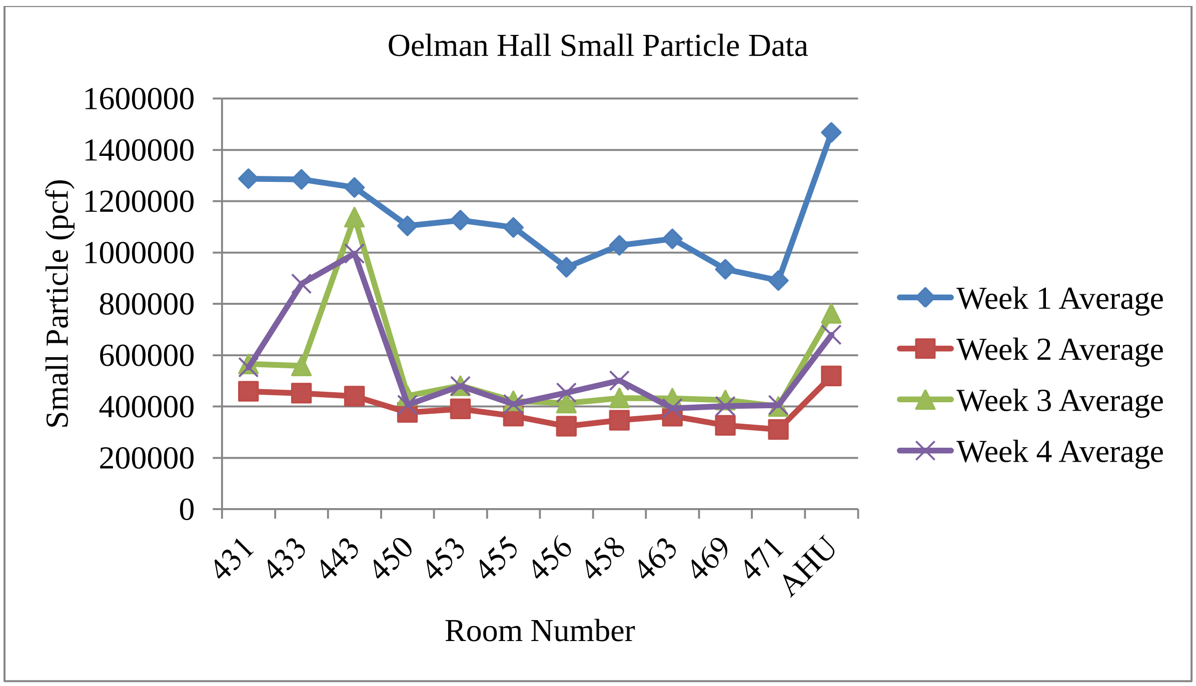

Figure 22. Oelman Hall Average Small Particle Concentrations

Similar to small particles, TVOCs do not have an overall standard for permissible indoor exposure limits; however, it is accepted that anything over 3 milligrams per meter cubed (mg/m^3) is considered hazardous with probable exposure effects (Fike, 2011). The sensor data for the WSU laboratories is reported in ppm as isobutylene, which requires a conversion for comparison. The conversion equation is given in Equation 13 and

assumes standard temperature and pressure. The molecular weight of isobutylene is 56.108 and 24.45 is a conversion factor representing the volume of one mole of gas (OSHA, 2014; SKC, 2014).

$$ppm\ value = \frac{\left(\frac{mg}{m^3}\ value\right)(24.45)}{56.108} \tag{13}$$

Using Equation 13 the TVOC hazardous level is 1.3 ppm as isobutylene. Figure 23 shows the average TVOC level maintained by room by week for Biological Science I Laboratory during the research period. There were four rooms that experience maximum TVOC readings greater than the accepted hazardous level; however, each event lasted no longer than 36 minutes and most were reduced to acceptable levels within 12 minutes.

Diggs Laboratory also experienced rooms with maximum TVOC levels greater than the threshold; however, it is unknown what was occurring in the space at the time of the event. For example, if an experiment was being conducted which generated VOCs or if VOCs were spilled in the lab a spike would be registered by the system. Yet, the system quickly returned the IAQ to acceptable levels. Figure 24 shows the average TVOC level for Diggs Laboratory.

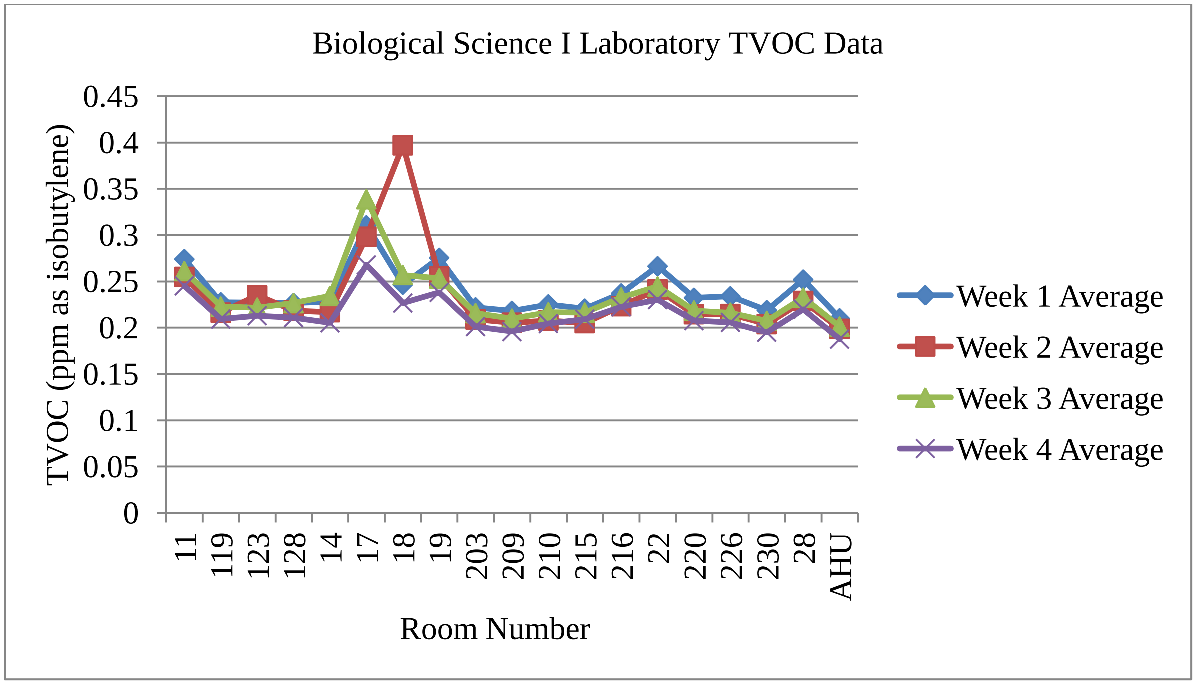

Figure 23. Biological Science I Laboratory Average TVOC Concentrations

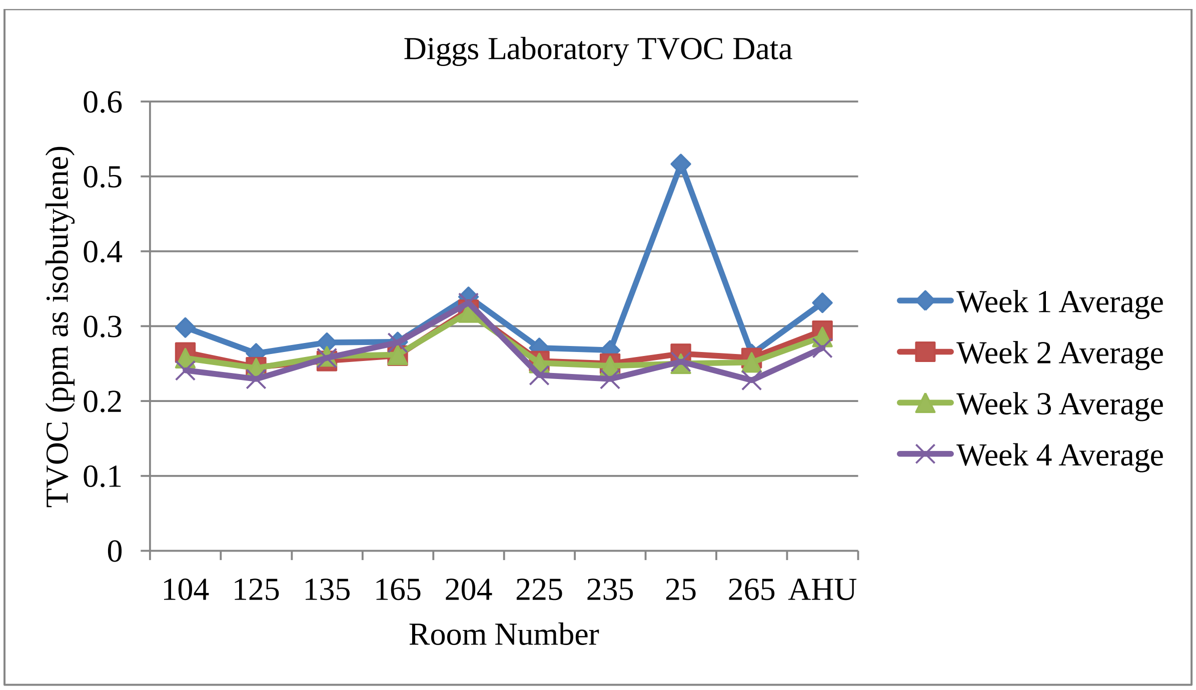

Figure 24. Diggs Laboratory Average TVOC Concentrations

Oelman Hall did not have any rooms experience a maximum TVOC level greater than 0.91 ppm as isobutylene. Figure 25 shows the average TVOC level by room for Oelman Hall. As shown, there is minimal change in TVOC concentrations from room to room and less than 0.1 ppm as isobutylene change from week to week.

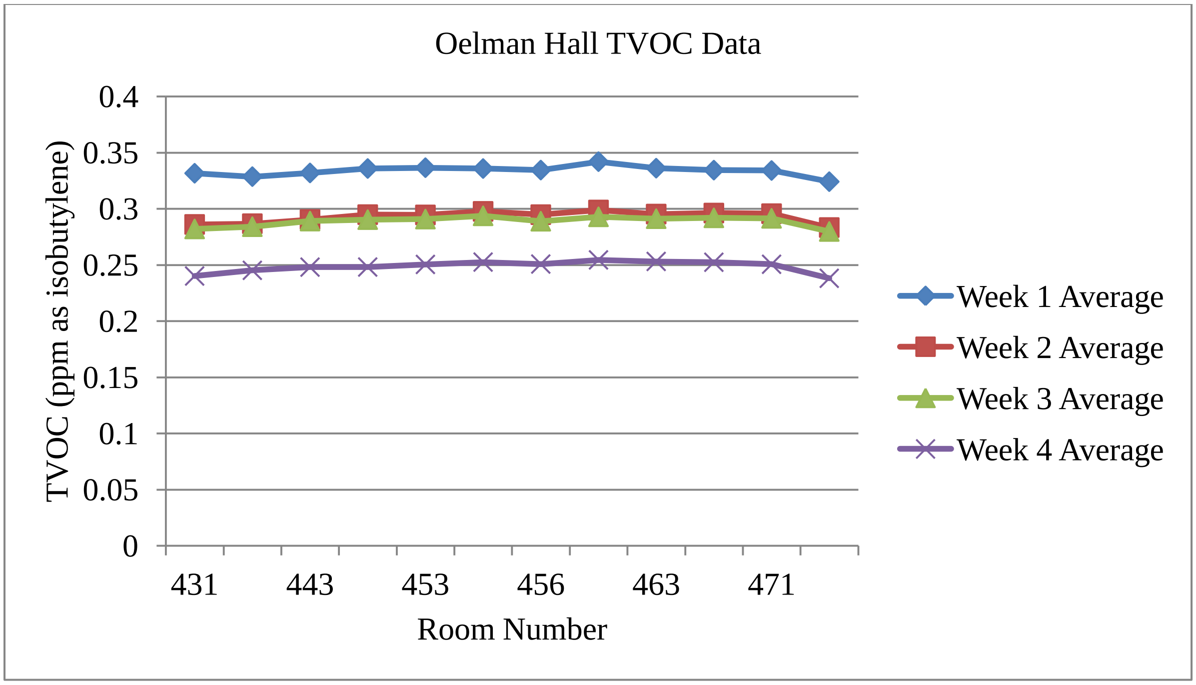

Figure 25. Oelman Hall Average TVOC Concentrations

With the exception of Bio Science I room 18 during the second week of the research period and Diggs Laboratory room 25 during the first week of the research period, the average TVOC concentration does not exceed 0.35 ppm as isobutylene (0.8 mg/m^3). This average is well within accepted levels for safety.

A DCV system enables the laboratory ACH rate to be reduced to meet the specific needs of the space and not just the design conditions. The DCV system results for CO_2,

small particles, and TVOC concentrations show that the DCV system is able to maintain the IAQ of laboratories at safe levels when the ACH baseline is reduced. Thus, the DCV system should be considered as a method of achieving energy savings in a laboratory.

Phase II

As discussed in the methodology chapter, Air Force laboratories range from 800 sq ft to 3000 sq ft in size with 12-foot high ceilings. Additionally, a conservative occupancy rate for Air Force laboratories is ten occupants per 1000 sq ft. The Battlespace Environment Laboratory (BEL) is a 146,300 sq ft facility that has eight laboratory zones; therefore, this analysis assumed that a typical USAF laboratory has from five to ten laboratory zones per facility. Table 9 provides a range of values determined by multiplying the considered square footage per zone by the considered number of zones. The smallest laboratory facility is 4000 sq ft with 40 occupants and the largest is 30,000 sq ft with 300 occupants. Only the values for 5 and 10 zones are shown in Table 9. The analysis considered facility sizes throughout the entire range in 500 sq ft increments.

Based on the total square footage, the minimum ventilation required is 0.9 ACH when the room is unoccupied; however, based on the maximum purge rate of 15 ACH established in Chapter III, the minimum flow rate that can be supplied by a variable frequency drive (VFD) is 3 ACH, or 20% of the maximum flow. Therefore, the minimum baseline is 3 ACH. When the room is occupied, the sensors will detect the occupancy and modulate the ventilation accordingly. The volume of ventilation required changes based on occupancy and square footage but the ACH rate remains constant

because increases in square footage and flow are linearly related. This range of total square footages was then used to calculate fan energy consumption.

Table 9. Range of USAF Laboratory Square Footage and Occupancy

sq ft /Zone	5 Zones		10 Zones	
	sq ft	Occupants	sq ft	Occupants
800	4000	40	8000	80
900	4500	45	9000	90
1000	5000	50	10000	100
1100	5500	55	11000	110
1200	6000	60	12000	120
1300	6500	65	13000	130
1400	7000	70	14000	140
1500	7500	75	15000	150
1600	8000	80	16000	160
1700	8500	85	17000	170
1800	9000	90	18000	180
1900	9500	95	19000	190
2000	10000	100	20000	200
2100	10500	105	21000	210
2200	11000	110	22000	220
2300	11500	115	23000	230
2400	12000	120	24000	240
2500	12500	125	25000	250
2600	13000	130	26000	260
2700	13500	135	27000	270
2800	14000	140	28000	280
2900	14500	145	29000	290
3000	15000	150	30000	300

Status Quo Energy Demand

The status quo condition is that the DOAS unit is providing eight ACH of ventilation to each laboratory zone throughout the entire year. The facility size ranges

from 4000 sq ft to 30,000 sq ft by 500 sq ft increments and the system total pressure ranges from 1 in. w.g. to 6 in. w.g. by 0.5 in. w.g. increments. Table 10 shows select combinations while Table 20 in Appendix B shows the annual fan energy consumption (in kWh) throughout the entire range of considered square footage and total pressure combinations. At a cost of $0.06/kWh, the range in the annual fan energy consumption shown in Table 10 ranges from $1,005 to $57,536(DOE: EERE, 2013).

Table 10. Status Quo Annual Fan Energy Consumption

Status Quo (8 ACH) Total Annual Fan Energy Consumption (kWh)						
Total Sq Ft	Total Pressure @ Full Flow (Supply/Exhaust)					
	1.0/0.5	2.0/1.5	3.0/2.5	4.0/3.5	5.0/4.5	6.0/5.5
4000	16747	39135	61522	83910	106297	128685
6000	25077	58572	92066	125561	159055	192550
8000	33408	78009	122611	167212	211814	256415
10000	41738	97446	153155	208863	264572	320280
12000	50068	116884	183699	250514	317330	384145
14000	58398	136321	214243	292166	370088	448010
16000	66729	155758	244787	333817	422846	511876
18000	75059	175195	275332	375468	475604	575741
20000	83389	194632	305876	417119	528362	639606
22000	91719	214070	336420	458770	581121	703471
24000	100050	233507	366964	500421	633879	767336
26000	108380	252944	397508	542073	686637	831201
28000	116710	272381	428053	583724	739395	895066
30000	125040	291819	458597	625375	792153	958931

As previously discussed the fan energy is approximately 85% of the total parasitic energy and the parasitic energy is approximately one-third of the total energy consumed in an HVAC system. Therefore, to determine overall HVAC energy consumption the fan

energy is divided by 85% and then multiplied by three. Table 11 shows the total annual HVAC energy consumption for the selected combinations. These energy consumption values were used in the economic analysis reported in the following section.

Table 11. Status Quo Annual HVAC Energy Consumption

Status Quo (8 ACH) Total Annual HVAC Energy Consumption (kWh)						
Total Sq Ft	Total Pressure @ Full Flow (Supply/Exhaust)					
	1.0/0.5	2.0/1.5	3.0/2.5	4.0/3.5	5.0/4.5	6.0/5.5
4000	59108	138123	217137	296152	375167	454182
6000	88509	206725	324940	443156	561372	679588
8000	117910	275326	432743	590160	747577	904994
10000	147310	343928	540546	737164	933782	1130400
12000	176711	412530	648349	884169	1119988	1355807
14000	206112	481132	756152	1031173	1306193	1581213
16000	235513	549734	863956	1178177	1492398	1806619
18000	264914	618336	971759	1325181	1678603	2032026
20000	294315	686938	1079562	1472185	1864809	2257432
22000	323715	755540	1187365	1619189	2051014	2482838
24000	353116	824142	1295168	1766193	2237219	2708245
26000	382517	892744	1402971	1913197	2423424	2933651
28000	411918	961346	1510774	2060202	2609630	3159057
30000	441319	1029948	1618577	2207206	2795835	3384464

DCV System Energy Demand

The DCV system uses sensors to monitor the IAQ and modulate the ventilation being supplied to the space based on demand. The baseline ventilation for this analysis is established by the maximum system purge rate and equipment limitations at 3 ACH even though according to the code the minimum ventilation required is 0.9 ACH. The system detects when additional ventilation is required and modulates rate it is supplied to the space as necessary. The average frequency, duration, and intensity of HVAC events was

determined in phase I of the methodology. It was determined that an average week requires the DCV system to respond to 7.1 events lasting 18.44 minutes at an intensity of 315.82 cfm above the baseline. Annually, the DCV system is above the baseline for 113.47 hours or 1.3% of the year.

Table 12 reports the DCV system annual supply air fan energy consumption for selected combinations of square footage and total pressure. Five to ten zones are considered in this analysis and the number of zones changes the fan energy consumption even if the square footage and static pressures are the same. Table 12 reports the average value when the square footages are the same for multiple different numbers of zones. Table 21 in Appendix B shows the annual fan energy consumption based on the zone average, when applicable, for the entire range of square footages and total pressures considered. At the same electricity cost of $0.06/kWh, the cost for annual energy consumption shown in Table 12 ranges from $70 to $3,939. The fan energy cost is greatly reduced when using the DCV system. Further, the cost range based on the total pressure and square footage is much tighter when compared to the status quo system.

Table 12. DCV Annual Fan Energy Consumption

DCV (Zones Averaged) Total Annual Fan Energy Consumption (kWh)						
Total Sq Ft	Total Pressure @ Full Flow (Supply/Exhaust)					
	1.0/0.5	2.0/1.5	3.0/2.5	4.0/3.5	5.0/4.5	6.0/5.5
4000	1163	2717	4270	5824	7377	8931
6000	1736	4050	6363	8676	10989	13303
8000	2309	5381	8453	11526	14598	17670
10000	2880	6710	10540	14370	18200	22030
12000	3451	8039	12626	17214	21801	26389
14000	4022	9368	14713	20058	25403	30748
16000	4594	10698	16801	22905	29008	35112
18000	5165	12027	18888	25749	32610	39471
20000	5737	13357	20976	28596	36215	43835
22000	6309	14687	23065	31442	39820	48198
24000	6880	16016	25151	34287	43422	52557
26000	7452	17346	27240	37133	47027	56921
28000	8024	18676	29328	39980	50632	61284
30000	8595	20005	31415	42824	54234	65644

The overall HVAC demand when using the DCV system was calculated following the same procedures for the status quo calculations. Table 13 presents the overall HVAC energy consumption for the DCV system. It is readily apparent that the DCV system generates substantial energy savings when compared to the status quo strategy throughout the range of considered conditions.

Table 13. DCV Annual HVAC Energy Consumption

DCV (Zones Averaged) Total Annual HVAC Energy Consumption (kWh)						
Total Sq Ft	Total Pressure @ Full Flow (Supply/Exhaust)					
	1.0/0.5	2.0/1.5	3.0/2.5	4.0/3.5	5.0/4.5	6.0/5.5
4000	4106	9589	15072	20555	26038	31520
6000	6128	14292	22457	30621	38786	46950
8000	8148	18991	29835	40678	51522	62365
10000	10164	23682	37199	50716	64234	77751
12000	12180	28372	44563	60754	76946	93137
14000	14197	33062	51927	70793	89658	108523
16000	16215	37757	59298	80840	102382	123924
18000	18231	42447	66662	90878	115094	139310
20000	20249	47141	74034	100926	127818	154710
22000	22267	51836	81405	110973	140542	170111
24000	24283	56526	88769	121012	153254	185497
26000	26301	61221	96140	131059	165978	200898
28000	28319	65915	103511	141107	178702	216298
30000	30336	70605	110875	151145	191414	231684

Phase III

Phase III synthesizes the results of phases I and II to compare the status quo and DCV laboratory HVAC energy consumption. Based on a savings-to-investment ratio (SIR) equal to one, the maximum initial DCV system cost is determined for each total pressure and square footage combination considered. An economic analysis was then performed for two conditions to determine the savings generated by the DCV system.

Status Quo and DCV Energy Consumption Comparison

In the previous section, Tables 10 and 12 report the fan energy consumption for the status quo and DCV conditions through a range of square footages and total pressures. Table 14 reports the savings achieved by the DCV system over the status quo

condition solely based on the fan energy savings. As shown, the amount of savings achieved by employing a DCV system depends greatly on the square footage being supplied and the total pressure. Using an electricity cost of $0.06/kWh, the annual fan energy savings using DCV range from $936 to $53,598.

Table 14. Annual Fan Energy Savings Achieved Using DCV

Total Annual Fan Energy Savings Using DCV (kWh)						
Total Sq Ft	Total Pressure @ Full Flow (Supply/Exhaust)					
	1.0/0.5	2.0/1.5	3.0/2.5	4.0/3.5	5.0/4.5	6.0/5.5
4000	15584	36418	57252	78086	98920	119754
6000	23341	54522	85704	116885	148066	179247
8000	31099	72628	114157	155687	197216	238745
10000	38858	90737	142615	194494	246372	298251
12000	46617	108845	171073	233301	295529	357756
14000	54376	126953	199530	272108	344685	417262
16000	62134	145060	227986	310912	393838	476764
18000	69893	163169	256444	349719	442994	536270
20000	77652	181276	284900	388523	492147	595771
22000	85410	199383	313355	427328	541300	655273
24000	93169	217491	341813	466135	590457	714779
26000	100928	235598	370269	504939	639610	774280
28000	108686	253705	398724	543744	688763	833782
30000	116445	271814	427182	582551	737919	893288

The total annual HVAC energy saved by the DCV system is reported in Table 15. The total annual HVAC savings achieved range from $189,167 to $3,300 annually. This range indicates that the economic feasibility of employing a DCV system depends significantly on the square footage of the facility and the total pressure of the HVAC system. However, remember that the conditioning energy is not dependent on the total pressure in the system. The following section uses the values reported in Table 15 in the

BLCC5 software program to determine whether the DCV system is life-cycle cost effective.

Table 15. Annual HVAC Energy Savings Achieved with DCV System

Total Annual HVAC Energy Savings Using DCV (kWh)						
Total Sq Ft	Total Pressure @ Full Flow (Supply/Exhaust)					
	1.0/0.5	2.0/1.5	3.0/2.5	4.0/3.5	5.0/4.5	6.0/5.5
4000	55002	128534	202066	275597	349129	422661
6000	82381	192432	302484	412535	522586	632638
8000	109762	256335	402909	549482	696056	842629
10000	137146	320247	503347	686448	869549	1052649
12000	164531	384159	603786	823414	1043042	1262670
14000	191915	448070	704225	960380	1216535	1472690
16000	219298	511978	804657	1097337	1390016	1682696
18000	246683	575889	905096	1234303	1563509	1892716
20000	274066	639797	1005528	1371259	1736990	2102722
22000	301448	703704	1105960	1508216	1910472	2312727
24000	328833	767616	1206399	1645182	2083965	2522748
26000	356216	831523	1306831	1782138	2257446	2732754
28000	383598	895431	1407263	1919095	2430927	2942759
30000	410983	959342	1507702	2056061	2604420	3152780

A sensitivity analysis was conducted for both fan efficiency and utility rates. These factors were chosen to determine the impact of fan selection and local utility rates on the savings potential of a DCV system. Figure 26 shows the results of the sensitivity analysis for three different facility sizes 10000 sq ft, 20000 sq ft, and 30000 sq ft and three different total pressures at full flow 1.0 in. w.g., 3.0 in. w.g., and 6.0 in. w.g. for the supply side. As shown, the impact of fan efficiency increases as the square footage of the facility being supplied increases or as the total pressure in the system increases. The most significant impact is for the 30000 sq ft facility at 6.0 in. w.g. which ranges from

91901 kWh to 51056 kWh. At $0.06 per kWh this difference costs $2,451 which is minimal when compared to total savings.

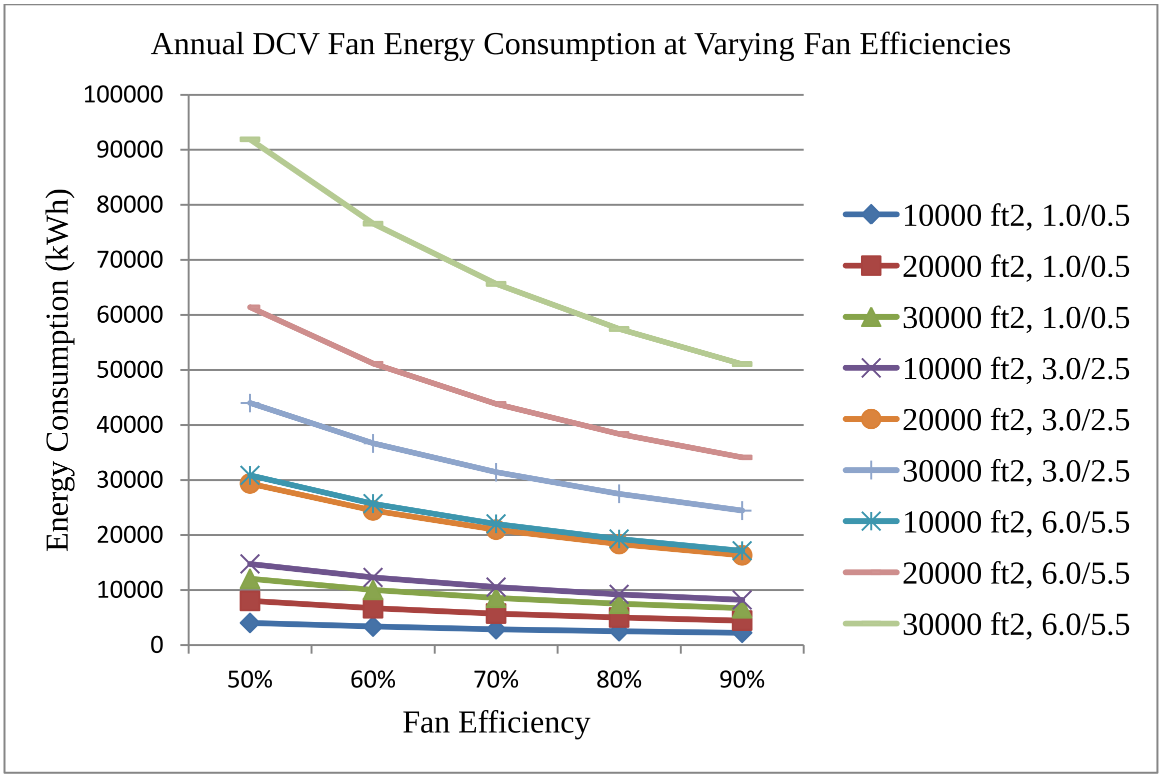

Figure 26. Annual DCV Fan Energy Consumption at Varying Fan Efficiencies

Figure 27 shows the sensitivity of the annual HVAC savings to changes in the local utility rate. The analysis considers electricity prices from $0.03 to $0.21 per kWh. Again, the impact of the change increases as the square footage of the facility increases and as the total pressure in the system increases. System with a total pressure of 1.0 in. w.g. on the supply side range less than $100,000 for the considered utility rates while system with a total pressure of 6.0 in. w.g. on the supply side have savings that range more than $500,000.

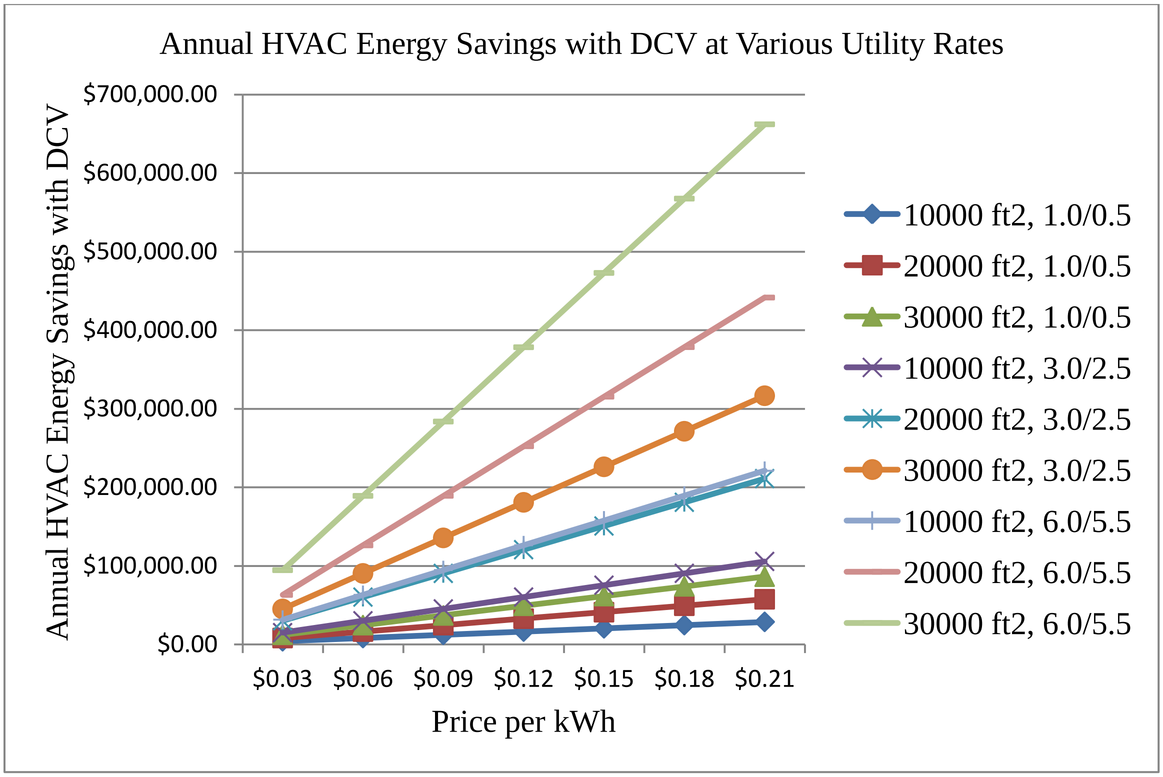

Figure 27. Annual HVAC Energy Savings with DCV at Various Utility rates

BLCC5 Results

The BLCC5 program used the HVAC energy consumption values reported in Table 15 to determine the initial system cost at which a SIR of 1 is achieved. This is the maximum price for the system to remain economically viable. Any system price over what is reported in Table 16 for the given conditions is not considered life-cycle cost effective. For less than 6000 sq ft with a static pressure of 1.0 in. w.g., there are insufficient savings achieved for the DCV system to ever be economically viable regardless of the initial system cost.

Table 16. Maximum DCV System Cost for SIR = 1

Maximum System Initial Cost for SIR = 1						
Total Sq ft	Total Pressure (Supply/Exhaust)					
	1.0/0.5	2.0/1.5	3.0/2.5	4.0/3.5	5.0/4.5	6.0/5.5
4000	($40,253)	$27,884	$96,021	$164,157	$232,293	$300,430
6000	($14,833)	$87,094	$189,071	$291,047	$393,024	$495,001
8000	$10,489	$146,308	$282,128	$417,946	$553,766	$689,584
10000	$35,864	$205,531	$375,196	$544,863	$714,529	$884,194
12000	$61,240	$264,753	$468,226	$671,779	$875,292	$1,078,806
14000	$86,615	$323,975	$561,335	$798,695	$1,036,056	$1,273,416
16000	$111,989	$383,194	$654,398	$925,604	$1,196,808	$1,468,013
18000	$137,364	$442,416	$747,468	$1,052,520	$1,357,571	$1,662,624
20000	$162,738	$501,634	$840,531	$1,179,427	$1,518,324	$1,857,221
22000	$188,111	$560,852	$933,594	$1,306,335	$1,679,077	$2,051,817
24000	$213,487	$620,075	$1,026,663	$1,433,252	$1,839,840	$2,246,428
26000	$238,861	$679,293	$1,119,726	$1,560,159	$2,000,592	$2,441,026
28000	$264,233	$738,512	$1,212,789	$1,687,067	$2,161,345	$2,635,622
30000	$289,609	$797,734	$1,305,859	$1,813,984	$2,322,108	$2,830,233

Table 17 reports the maximum system initial cost per square foot in order to achieve a SIR of one. The cost of the WSU laboratory DCV systems was approximately $8/sq ft for the laboratories over 10,000 sq ft (Diggs Laboratory and Biological Science I). The cost per square foot for Oelhman Hall (4,992 sq ft) was just over $21/sq ft. This range in cost per square foot is due to the DCV system function and number of zones. The DCV system installed uses centrally located pumps and sensor suites to extract and analyze air from each space. Regardless of square footage, each DCV system has a minimum installation cost. Additionally, Oelman Hall has 11 different laboratory zones that are being monitored while Diggs Laboratory has only nine, and additional system hardware is required to support each additional zone. Using these costs as a reference,

facilities less than 10,000 sq ft the maximum cost per sq ft should be greater than $21 and for facilities 10,000 sq ft and greater the maximum cost per sq ft should be greater than $8, the conditions whose values are shaded are not cost effective.

Table 17. Maximum DCV System Cost/sq ft for SIR = 1

Maximum DCV System Cost/Sq ft for SIR = 1						
Total Sq ft	Total Pressure (Supply/Exhaust)					
	1.0/0.5	2.0/1.5	3.0/2.5	4.0/3.5	5.0/4.5	6.0/5.5
4000	---	$6.97	$24.01	$41.04	$58.07	$75.11
6000	---	$14.52	$31.51	$48.51	$65.50	$82.50
8000	$1.31	$18.29	$35.27	$52.24	$69.22	$86.20
10000	$3.59	$20.55	$37.52	$54.49	$71.45	$88.42
12000	$5.10	$22.06	$39.02	$55.98	$72.94	$89.90
14000	$6.19	$23.14	$40.10	$57.05	$74.00	$90.96
16000	$7.00	$23.95	$40.90	$57.85	$74.80	$91.75
18000	$7.63	$24.58	$41.53	$58.47	$75.42	$92.37
20000	$8.14	$25.08	$42.03	$58.97	$75.92	$92.86
22000	$8.55	$25.49	$42.44	$59.38	$76.32	$93.26
24000	$8.90	$25.84	$42.78	$59.72	$76.66	$93.60
26000	$9.19	$26.13	$43.07	$60.01	$76.95	$93.89
28000	$9.44	$26.38	$43.31	$60.25	$77.19	$94.13
30000	$9.65	$26.59	$43.53	$60.47	$77.40	$94.34

Tables 16 and 17 show for which conditions the DCV system is able to be cost effective and the maximum system price. System total pressure is paramount to determining the potential savings of a DCV system and if the supply system total pressure is greater than 2.0 in. w.g., the DCV system is life-cycle cost effective when using WSU laboratory system costs.

A more detailed analysis is provided for a 10,000 and 20,000 sq ft facility with a supply side total pressure of 2.0 in. w.g. and an exhaust side total pressure of 1.5 in. w.g. to determine actual cost savings. Also considered is a 10,000 sq ft facility with a supply side total pressure of 5.0 in. w.g. and an exhaust side total pressure of 4.5 in. w.g. For these facilities, the initial system cost is assumed to be $8/sq ft in accordance with WSU laboratory DCV costs. At this price, the DCV system initial cost is $80,000 and $160,000, respectively. The DCV system on the 10,000 sq ft facility at the lower total pressures yields a SIR of 2.57 and an adjusted internal rate of return (AIRR) of 7.98%. Since the SIR is above one and the AIRR is above the mandated 3% for energy savings projects, this facility is a candidate for DCV implementation. The simple payback occurs in the 5th year and the discounted payback occurs in the 6th year. Additionally, the net savings is $125,531 by saving more than 6,404 MWh of electricity over the life of the system. A DCV system implemented on the 20,000 sq ft facility and the described total pressure yields a SIR of 3.14 and an AIRR of 9.06%. The simple payback and the discounted payback occur in the 5th year. The total net savings is $341,634 by saving more than 12,794 MWh of electricity over the life of the system. For the different size facilities the strength of the investment varies but the payback occurs in approximately the same amount of time.

The last in-depth analysis is for a 10,000 sq ft facility with supply and exhaust total pressure that is 3.0 in. w.g. higher than previously analyzed. For these conditions the DCV achieves a SIR of 8.93 and an AIRR of 14.92%. Both the simple and the discounted payback occur in the 2nd year. The net savings is $634,529 compared to the status quo and the total energy saved during the life of the system is 17,388 MWh. An

increase in 10,000 sq ft, at the same total pressure, saves an additional$216,103 in net savings. An increase of 3.0 in. w.g., at the same square footage, saves an additional $508,998 in net savings. The square footage and system pressure both impact the potential savings of a DCV system to varying degrees which makes it paramount to determine potential savings based on the actual system characteristics.

These results provide a quick method to determine if further investigation into DCV implementation is warranted. In each of these scenarios the DCV system is able to yield a SIR that is greater than one. However, the strength of each facility for DCV implementation varies with the facility characteristics.

Summary

The results from each phase of this research effort were presented to determine that an average laboratory DCV system engages to increase airflow more than 50 cfm above the baseline 1.3% of the time. Further, the DCV system is able to maintain a safe IAQ at the reduced baseline. The fan savings generated from the DCV system vary greatly depending on the total system pressure and square footage of the space being supplied. The HVAC energy savings will also vary based on the system and also local utility rates and location. However, a DCV strategy for a DOAS is life-cycle cost effective for the majority of conditions considered.

V. Conclusions and Recommendations

Chapter Overview

This chapter reviews the findings of this research and discusses their significance. The chapter begins with a recommendation on how to use the results of this thesis as an initial screening tool for energy savings projects. The limitations encountered in the research are then explained. Lastly, the future research possibilities resulting from this research is discussed.

Review of Findings

In the first chapter, the primary research question and three investigative questions were posed. The primary research question asked if a life-cycle cost effective carbon dioxide (CO_2)-based demand controlled ventilation (DCV) strategy could be used to reduce energy demand for laboratory facilities while maintaining the recommended indoor air quality (IAQ). Chapter III outlined the methodology used to answer this question, and Chapter IV presented the results of the methodology to answer the primary research question.

The three investigative questions focused the research and formed the foundation for answering the primary research question. The investigative questions sought to determine the amount of energy reduced, how IAQ was affected, and the costs saved as a result of implementing a CO_2-based DCV ventilation strategy. Each of these questions was answered in Chapter IV. The first investigative question was answered in phase II which presented the amount of fan and heating, ventilating, and air-conditioning (HVAC) energy savings results from a DCV system. The second investigative question was also answered in phase II by examining the IAQ of the Wright State University (WSU)

laboratories. It was shown that a DCV system is able to maintain an acceptable IAQ in a laboratory even though the baseline air change rate is reduced. The final investigative question was answered in phase III with the results from the BLCC5 software. The savings potential of a DCV system is highly dependent on the square footage of the facility and the total pressure in the HVAC system.

Significance of Research

With the recent update to Unified Facilities Criteria (UFC) 3-410-01 in July of 2013 new Air Force laboratory facility construction will be required to use a dedicated outdoor air system (DOAS) when the ventilation requirement exceeds 1000 cubic feet per minute (cfm). While the UFC is specific to the military, Roth et al. (2002) identified DOAS as one of the top 15 HVAC energy savings opportunities and asserts that a DOAS has “superior humidity control” (Roth et al., 2002, 4-7). Improved humidity control helps HVAC designers to achieve setpoints and control limits while saving energy, which is integral in laboratory design. This research shows that under many laboratory conditions, a DOAS can be coupled with a DCV system to achieve energy savings. The laboratory facility managers can use the results presented as a quick screening tool to determine if a laboratory should be considered for DCV.

Additionally, this research shows that a DCV system can maintain acceptable IAQ in a laboratory setting at a reduced air change rate baseline. Therefore, safety is not sacrificed to achieve energy savings. Based on this knowledge it would be advantageous to investigate and potentially modify existing laboratory ventilation practices. If the IAQ of the laboratory space was monitored, the safety of the laboratory environment can be tracked and ventilation can be reduced.

Limitations

There are several limitations to the application of this research effort to determine laboratory energy savings using DCV. Each limitation is accounted for in this research; however, the generalizations used should be more accurately defined when determining if a DCV system is appropriate. The primary limitation is the HVAC system setup while secondary limitations include location specific requirements and laboratory setpoint control limits.

Future Air Force laboratories will employ a DOAS in parallel with another system to supply air to laboratories as efficiently as possible; however, this requirement was only recently incorporated into UFC 3-410-01. Many existing laboratories do not use a parallel DOAS for ventilation; thus, these facilities cannot use the results of this research. However, these facilities may still benefit from DCV implementation based on the success of DCV at WSU. The determination to implement DCV on these facilities will require a location and system specific analysis.

This research effort does not consider the varying HVAC requirements based on facility location. The climate can significantly impact and alter HVAC operation. Specifically, a humid climate requires that the HVAC system maintain the building envelope at a positive pressure relative to outdoor conditions to prevent infiltration. Additionally, this climate requires a significant focus on dehumidification to avoid moisture buildup and mold growth (MacPhaul & Etter, 2010). Conversely, a cold dry climate may require that the HVAC system add humidity to the supply air (Miles & Furgeson, 2008). Furthermore, the demand for heating and cooling is drastically

different in each of these climates. Based on the specific HVAC requirements, the ratio of heating and cooling to parasitic energy consumption will vary.

Lastly, the work being performed in a laboratory may require tight control limits on the HVAC setpoints (e.g., temperature, percent relative humidity, etc.). These control limits can limit the possible energy reduction because of their impact on system operation. The system may not be able to reduce airflow due to the airflow required to maintain the desired setpoints. In this analysis, it was assumed that the parallel system was able to maintain desired setpoints in conjunction with the DOAS operation. This assumption may not hold true depending on the climate, setpoint, and corresponding limits for that setpoint.

Recommendations for Action

The results from this research effort are to be used as a quick screening tool to determine candidate laboratory facilities for the implementation of a DCV system. Laboratories matching the conditions for cost effective systems in Table 17 should be considered for DCV system implementation to achieve energy savings. Additionally, any new laboratories being planned should consult Table 16 to determine if the laboratory is a good candidate for DCV implementation. However, just because a laboratory meets the conditions presented does not automatically mean that the laboratory will achieve savings. A location specific in-depth analysis needs to be completed to more accurately predict the potential savings from a DCV system.

Recommendations for Future Research

This research effort made some assumptions and had some limitations that can be explored through future research. It was assumed that the parasitic energy was one-third

of the total HVAC energy requirements. This assumption can be validated by modeling the HVAC energy requirements specific to laboratories for different climates. Additionally, the HVAC setup analyzed was a DOAS in parallel with another HVAC system. The application of DCV can be expanded by determining the effect of DCV in a laboratory using different HVAC setups. Lastly, a different facility type with similar 100% outdoor air requirements can be investigated to determine the effect of DCV implementation.

An energy model specifically designed to determine energy savings from DCV installed in a laboratory with a DOAS will provide more accurate location specific energy savings. As discussed earlier, the climate of a location can drive different requirements for the HVAC system which has impacts on HVAC operation. An energy model will be able to more accurately assess the heating and cooling energy associated with HVAC operation for a specific location. A document produced by the U.S. Department of Energy (DOE) asserts that DCV generates greater savings in colder climates by reducing the demand for heating (DOE, 2012). However, DOAS is becoming increasingly popular to handle the high latent loads found in warm humid climates due to its improved efficiency (Larrañaga, Beruvides, Holder, Karunasena, & Straus, 2008). This research would provide a screening tool for climates where a DOAS using DCV will generate the greatest savings.

Many of the existing HVAC systems serving laboratories do not function in parallel with a DOAS. For the varying system types currently in use, would a DCV system be able to achieve energy savings? Additionally, how much savings would the DCV system be able to generate? The laboratories at WSU do not function with a

parallel DOAS unit and were able to achieving significant energy savings. This type of research effort would be able to identify existing laboratory HVAC system configurations most likely to benefit for a DCV system. Furthermore, the analysis would provide an initial estimate for the potential savings.

Hospitals are similar to laboratories because certain areas of the facility may require 100% outdoor air. This requirement increases energy consumption and makes hospitals a candidate for DCV for the same reason that many laboratories are a good candidate for DCV. The National Renewable Energy Laboratoy (NREL) published a technical report documenting the use of both DOAS and DCV in a large hospital to achieve energy savings. DCV was applied specifically to areas where occupancy determined the ventilation requirement (NREL, 2010). Another research effort could investigate the effect of using DCV throughout the entire hospital by reducing the baseline ventilation requirement and monitoring the IAQ of the space.

Summary

This effort sought to determine how a life-cycle cost effective DCV system could be implemented in laboratories to achieve energy savings while maintaining the IAQ. For a range of square footages and total pressure, a DCV system can be implemented to achieve net savings without adversely affecting IAQ. This effort provides a tool for any laboratory facility manager to quickly determine if a laboratory is a good candidate for possible DCV implementation.

Appendix A

Table 18. Acronyms Quick Reference

Acronym	Explanation
ACGIH	American Conference of Industrial Hygienists
ACH	Air Changes per Hour
AHP	Air Horsepower
AHU	Air Handling Unit
ANSI	American National Standards Institute
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
BEL	Battlespace Engineering Laboratory
BLCC5	Building Life Cycle Cost Software
CAV	Constant Air Volume
CO_2	Carbon Dioxide
DCV	Demand Controlled Ventilation
DOAS	Dedicated Outdoor Air System
DoD	U.S. Department of Defense
DOE	U.S. Department of Energy
EERE	DOE Office of Energy Efficiency and Renewable Energy
FCU	Fan Coil Unit
HVAC	Heating, Ventilating, and Air-Conditioning
IAQ	Indoor Air Quality
IEA	International Energy Agency
IR	Infrared
LCCA	Life-Cycle Cost Analysis
MILCON	Military Construction
NIST	National Institute of Standards and Technology
NRC	National Research Council
NREL	National Renewable Energy Laboratory
OSHA	Occupational Safety and Health Administration
SBS	Sick Building Syndrome
SIR	Savings-to-Investment Ratio
TVOC	Total Volatile Organic Compound
UCI	University of California - Irvine
UFC	United Facilities Criteria
UNDP	United Nations Development Programme
USAF	United States Air Force
VAV	Variable Air Volume
VFD	Variable Frequency Drive
WSU	Wright State University

Table 19. Units Quick Reference

Unit	**Explanation**
cfm	Cubic feet per minute
fpm	Feet per minute
g/kg	Grams per kilogram
hp	Horsepower
in. w.g.	Inches water gauge
kW	Kilowatt
kWh	Kilowatt-hour
L/s	Liters per second
$L/s/m^2$	Liters per second per meters squared
m^2	Meters squared
m^3/hr	Meters cubed per hour
pcf	Particles per cubic foot
ppm	Parts per million
Sq ft	Square feet (also ft^2)

Appendix B

Table 20. Status Quo Annual Fan Energy Consumption (Complete)

Status Quo (8 ACH) Total Annual Fan Energy Consumption (kWh)											
Total Sq Ft	Total Pressure @ Full Flow (Supply/Exhaust)										
	1.0/0.5	1.5/1.0	2.0/1.5	2.5/2.0	3.0/2.5	3.5/3.0	4.0/3.5	4.5/4.0	5.0/4.5	5.5/5.0	6.0/5.5
4000	16747	27941	39135	50328	61522	72716	83910	95104	106297	117491	128685
4500	18830	31412	43994	56576	69158	81740	94323	106905	119487	132069	144651
5000	20912	34883	48853	62824	76794	90765	104735	118706	132676	146647	160617
5500	22995	38354	53713	69072	84430	99789	115148	130507	145866	161225	176584
6000	25077	41825	58572	75319	92066	108814	125561	142308	159055	175803	192550
6500	27160	45296	63431	81567	99702	117838	135974	154109	172245	190381	208516
7000	29243	48767	68291	87815	107339	126863	146387	165910	185434	204958	224482
7500	31325	52238	73150	94062	114975	135887	156799	177712	198624	219536	240449
8000	33408	55708	78009	100310	122611	144911	167212	189513	211814	234114	256415
8500	35490	59179	82868	106558	130247	153936	177625	201314	225003	248692	272381
9000	37573	62650	87728	112805	137883	162960	188038	213115	238193	263270	288348
9500	39655	66121	92587	119053	145519	171985	198450	224916	251382	277848	304314
10000	41738	69592	97446	125301	153155	181009	208863	236717	264572	292426	320280
10500	43821	73063	102306	131548	160791	190033	219276	248519	277761	307004	336246
11000	45903	76534	107165	137796	168427	199058	229689	260320	290951	321582	352213
11500	47986	80005	112024	144044	176063	208082	240102	272121	304140	336160	368179
12000	50068	83476	116884	150291	183699	217107	250514	283922	317330	350738	384145
12500	52151	86947	121743	156539	191335	226131	260927	295723	330519	365315	400112
13000	54233	90418	126602	162787	198971	235156	271340	307524	343709	379893	416078
13500	56316	93889	131462	169034	206607	244180	281753	319326	356898	394471	432044
14000	58398	97360	136321	175282	214243	253204	292166	331127	370088	409049	448010
14500	60481	100831	141180	181530	221879	262229	302578	342928	383278	423627	463977
15000	62564	104301	146039	187777	229515	271253	312991	354729	396467	438205	479943
15500	64646	107772	150899	194025	237151	280278	323404	366530	409657	452783	495909
16000	66729	111243	155758	200273	244787	289302	333817	378331	422846	467361	511876
16500	68811	114714	160617	206520	252423	298327	344230	390133	436036	481939	527842
17000	70894	118185	165477	212768	260059	307351	354642	401934	449225	496517	543808
17500	72976	121656	170336	219016	267696	316375	365055	413735	462415	511095	559774
18000	75059	125127	175195	225263	275332	325400	375468	425536	475604	525672	575741
18500	77141	128598	180055	231511	282968	334424	385881	437337	488794	540250	591707
19000	79224	132069	184914	237759	290604	343449	396294	449138	501983	554828	607673
19500	81307	135540	189773	244006	298240	352473	406706	460940	515173	569406	623639

20000	83389	139011	194632	250254	305876	361497	417119	472741	528362	583984	639606
20500	85472	142482	199492	256502	313512	370522	427532	484542	541552	598562	655572
21000	87554	145953	204351	262749	321148	379546	437945	496343	554742	613140	671538
21500	89637	149424	209210	268997	328784	388571	448357	508144	567931	627718	687505
22000	91719	152895	214070	275245	336420	397595	458770	519945	581121	642296	703471
22500	93802	156365	218929	281492	344056	406620	469183	531747	594310	656874	719437
23000	95884	159836	223788	287740	351692	415644	479596	543548	607500	671452	735403
23500	97967	163307	228648	293988	359328	424668	490009	555349	620689	686029	751370
24000	100050	166778	233507	300236	366964	433693	500421	567150	633879	700607	767336
24500	102132	170249	238366	306483	374600	442717	510834	578951	647068	715185	783302
25000	104215	173720	243225	312731	382236	451742	521247	590752	660258	729763	799269
25500	106297	177191	248085	318979	389872	460766	531660	602554	673447	744341	815235
26000	108380	180662	252944	325226	397508	469790	542073	614355	686637	758919	831201
26500	110462	184133	257803	331474	405144	478815	552485	626156	699826	773497	847167
27000	112545	187604	262663	337722	412780	487839	562898	637957	713016	788075	863134
27500	114627	191075	267522	343969	420416	496864	573311	649758	726205	802653	879100
28000	116710	194546	272381	350217	428053	505888	583724	661559	739395	817231	895066
28500	118793	198017	277241	356465	435689	514913	594137	673361	752585	831809	911033
29000	120875	201488	282100	362712	443325	523937	604549	685162	765774	846386	926999
29500	122958	204958	286959	368960	450961	532961	614962	696963	778964	860964	942965
30000	125040	208429	291819	375208	458597	541986	625375	708764	792153	875542	958931

Table 21. DCV (Zones Averaged) Annual Fan Energy Consumption (Complete)

DCV (Zones Averaged) Total Annual Fan Energy Consumption (kWh)											
Total Sq Ft	Total Pressure @ Full Flow (Supply/Exhaust)										
	1.0/0.5	1.5/1.0	2.0/1.5	2.5/2.0	3.0/2.5	3.5/3.0	4.0/3.5	4.5/4.0	5.0/4.5	5.5/5.0	6.0/5.5
4000	1163	1940	2717	3494	4270	5047	5824	6601	7377	8154	8931
4500	1306	2178	3049	3921	4792	5663	6535	7406	8278	9149	10021
5000	1450	2416	3383	4349	5316	6282	7249	8215	9182	10148	11115
5500	1593	2654	3716	4778	5839	6901	7962	9024	10085	11147	12209
6000	1736	2893	4050	5206	6363	7519	8676	9833	10989	12146	13303
6500	1879	3130	4382	5633	6884	8136	9387	10638	11890	13141	14392
7000	2022	3369	4715	6062	7408	8754	10101	11447	12794	14140	15486
7500	2165	3606	5047	6489	7930	9371	10812	12253	13694	15135	16576
8000	2309	3845	5381	6917	8453	9989	11526	13062	14598	16134	17670
8500	2451	4082	5713	7344	8975	10606	12237	13867	15498	17129	18760
9000	2594	4320	6045	7771	9496	11222	12948	14673	16399	18124	19850
9500	2737	4557	6378	8198	10018	11838	13659	15479	17299	19119	20940
10000	2880	4795	6710	8625	10540	12455	14370	16285	18200	20115	22030
10500	3023	5032	7042	9052	11061	13071	15081	17090	19100	21110	23119
11000	3165	5270	7374	9479	11583	13687	15792	17896	20000	22105	24209
11500	3308	5507	7706	9906	12105	14304	16503	18702	20901	23100	25299
12000	3451	5745	8039	10332	12626	14920	17214	19508	21801	24095	26389
12500	3594	5982	8371	10759	13148	15536	17925	20313	22702	25090	27479
13000	3737	6220	8703	11186	13669	16153	18636	21119	23602	26085	28569
13500	3880	6457	9035	11613	14191	16769	19347	21925	24503	27081	29658
14000	4022	6695	9368	12040	14713	17385	20058	22730	25403	28076	30748
14500	4165	6932	9700	12467	15234	18002	20769	23536	26304	29071	31838
15000	4308	7170	10032	12894	15756	18618	21480	24342	27204	30066	32928
15500	4451	7408	10365	13323	16280	19237	22194	25151	28108	31065	34022
16000	4594	7646	10698	13749	16801	19853	22905	25956	29008	32060	35112
16500	4737	7883	11030	14176	17323	20469	23616	26762	29909	33055	36202
17000	4880	8121	11362	14603	17844	21086	24327	27568	30809	34050	37291
17500	5023	8358	11694	15030	18366	21702	25038	28374	31710	35045	38381
18000	5165	8596	12027	15457	18888	22318	25749	29179	32610	36041	39471
18500	5309	8834	12360	15886	19411	22937	26463	29988	33514	37039	40565
19000	5452	9072	12692	16313	19933	23553	27174	30794	34414	38035	41655
19500	5594	9309	13024	16740	20455	24170	27885	31600	35315	39030	42745
20000	5737	9547	13357	17166	20976	24786	28596	32405	36215	40025	43835
20500	5880	9784	13689	17593	21498	25402	29307	33211	37116	41020	44924

21000	6023	10022	14021	18020	22019	26019	30018	34017	38016	42015	46014
21500	6166	10260	14355	18449	22543	26637	30731	34826	38920	43014	47108
22000	6309	10498	14687	18876	23065	27254	31442	35631	39820	44009	48198
22500	6452	10735	15019	19303	23586	27870	32154	36437	40721	45004	49288
23000	6595	10973	15351	19730	24108	28486	32865	37243	41621	45999	50378
23500	6737	11210	15683	20157	24630	29103	33576	38049	42522	46995	51468
24000	6880	11448	16016	20583	25151	29719	34287	38854	43422	47990	52557
24500	7024	11686	16349	21012	25675	30338	35000	39663	44326	48989	53651
25000	7166	11924	16681	21439	26196	30954	35711	40469	45226	49984	54741
25500	7309	12161	17014	21866	26718	31570	36422	41275	46127	50979	55831
26000	7452	12399	17346	22293	27240	32187	37133	42080	47027	51974	56921
26500	7595	12636	17678	22720	27761	32803	37844	42886	47928	52969	58011
27000	7738	12874	18010	23147	28283	33419	38555	43692	48828	53964	59101
27500	7881	13112	18344	23575	28806	34038	39269	44501	49732	54963	60195
28000	8024	13350	18676	24002	29328	34654	39980	45306	50632	55958	61284
28500	8167	13587	19008	24429	29850	35270	40691	46112	51533	56954	62374
29000	8309	13825	19340	24856	30371	35887	41402	46918	52433	57949	63464
29500	8452	14062	19673	25283	30893	36503	42113	47723	53334	58944	64554
30000	8595	14300	20005	25710	31415	37119	42824	48529	54234	59939	65644

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13. SUPPLEMENTARY NOTES

14. ABSTRACT
The building sector in the United States accounted for 41% of domestic and 7% of global energy consumption in 2010, with heating, ventilating, and air-conditioning (HVAC) activities consuming approximately 41.4% of the total facility energy consumption. Within the HVAC system, the parasitic energy accounts for one-third of the total energy consumed while heating and cooling accounts for the balance. The fan energy is approximately 85% of the total parasitic energy in the HVAC system. In a laboratory, energy related to ventilation can account for nearly half of the electrical energy demand. A carbon dioxide (CO2) – based demand controlled ventilation (DCV) strategy can reduce the ventilation requirement by monitoring the indoor air quality (IAQ) of a space and modulating the ventilation based on the real-time occupancy. This research presents a tool for laboratory managers to quickly determine if employing a DCV system is potentially life-cycle cost effective. The tool presented is not to be used as sole justification for implementing a DCV system; instead, laboratory managers using this tool will be able to quickly determine if further investigation into DCV installation is warranted. The results show that a DCV system is life-cycle cost effective for many different HVAC system total pressure and square footage combinations.

15. SUBJECT TERMS
HVAC, DCV, DOAS, Laboratory HVAC Energy

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